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N73 26895

SHUTTLE CRYOGENICS SUPPLY SYSTEM OPTIMIZATION STUDY. VOL. 4: TECH. REPORT CRYOGENIC COOLING IN ENVIRONMENTAL CONTROL SYSTEMS - FINAL REPORT

LOCKHEED MISSILES & SPACE COMPANY, INC.
SUNNYVALE, CA

JUN 73

CR-133952

SHUTTLE

LMSC-A991396
30 JUNE 1973

FINAL REPORT

SHUTTLE CRYOGENICS SUPPLY SYSTEM

OPTIMIZATION STUDY

(NASA-CR-133952) SHUTTLE CRYOGENIC SUPPLY SYSTEM OPTIMIZATION STUDY. VOLUME 4: CRYOGENIC COOLING IN ENVIRONMENTAL CONTROL SYSTEMS Final Report (Lockheed Missiles and Space Co.) 225 p HC $3.25 CSCL 22B

N73-26895

Unclas
G3/31 08742

CRYOGENIC COOLING
IN
ENVIRONMENTAL CONTROL SYSTEMS

CONTRACT NAS9-11330

Prepared for Manned Spacecraft Center
by
Manned Space Programs, Space Systems Division

LOCKHEED MISSILES & SPACE COMPANY, INC.
A SUBSIDIARY OF LOCKHEED AIRCRAFT CORPORATION

DRL Line Item 9
DRD No. MA-129T

IMSC-A991396

FINAL REPORT

SHUTTLE CRYOGENIC SUPPLY SYSTEM

OPTIMIZATION STUDY

VOLUME IV

TECHNICAL REPORT

CRYOGENIC COOLING IN ENVIRONMENTAL CONTROL SYSTEMS

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FOREWORD

This volume provides the results obtained in Task 1A – Cryogenic Cooling in Environmental Control System of the Shuttle Cryogenics Supply System Optimization Study, NAS 9-11330, performed by Lockheed Missiles & Space Company (LMSC) under contract to the National Aeronautics and Space Administration, Manned Spacecraft Center, Houston, Texas. The study was under the technical direction of Mr. T. L. Davies, Cryogenics Section of the Power Generation Branch, Propulsion and Power Division.

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Section 1

INTRODUCTION AND SUMMARY

1.1 INTRODUCTION

The Space Shuttle System was initiated by the National Aeronautics and Space Administration to provide a low-cost space transportation system, chiefly through the use of reusable vehicles. The system is to become operational in the period 1976 to 1980.

Vehicle configurations used during the Space Shuttle Phase B definition studies were two-stage fully reusable vehicles, consisting of a booster and an orbiter. The orbiter vehicle contained cryogenic fluid systems which supplied propellants for all propulsion system as well as reactants for the power-generation systems.

The Shuttle Cryogenic Supply System (SCSS) Optimization Study was initiated by NASA to determine the manner in which the cryogenic fluid storage and supply tanks and subsystems might be treated as integrated systems. One task of the overall study was to determine the feasibility and practicality of replacing or supplementing the Space Shuttle orbiter radiators with cryogenic cooling; this task was entitled Cryogenic Cooling in Environmental Control Systems, Task 1A. This report describes the studies and conclusions associated with the Task 1A effort, which was started approximately 9 months after the Shuttle Cryogenic Supply System Optimization Study was begun and which utilized many of the requirements, systems descriptions, and data generated therein.

During the course of the study, NASA redirected the configuration designs being studied by the Phase B and Alternate Concepts Studies contractors, and this redirection had a significant impact on the Task 1A studies. The shuttle configuration was modified from a two-stage fully reusable system consisting of a recoverable orbiter and booster to a recoverable orbiter with external tanks

and solid-rocket first stage. The orbiter changed from a configuration with internal cryogenic hydrogen and oxygen main propulsion tanks, and cryogenic oxygen and hydrogen for Orbit Maneuvering Propulsion (OMPS), the Reaction Control System (RCS), the fuel cell and APU reactants to a configuration with external expendable oxygen and hydrogen main propulsion tanks and earth-storable propellant for OMPS, RCS, and APU. The only cryogens remaining in the orbiter are for full cell operation and potentially for EC/LSS cooling.

1.2 PURPOSE

Initially the study was designed to determine ways in which the available large quantities of cryogens could be used to absorb the heat generated by the electronics and the crew and to utilize this heat beneficially to condition the cryogens for their ultimate use. It was anticipated that the orbiter radiators could be eliminated aor at least reduced in size. Furthermore, it was expected that if the radiators were not eliminated, the on-board cryogens would play a key roll in providing the cooling function at times when the radiators were not deployed.

The initial studies were begun with the just enumerated purposes in mind and in the following major categories:

- No work to be removed from the cryogens
- Sufficient work to be removed from the cryogens to power compressors or pumps
- As much work as practical to be removed from the cryogens to supplement vehicle power

General system concepts were defined to help evaluate these major categories and are:

- Expel cryogens overboard directly after absorbing heat
- Store heated cryogens in accumulators
- Store heated cryogens in ascent tanks
- Feed crogens direct to user after heating

Studies in these general areas were initiated and effort had proceeded for a few months when the design change was announced. At that time the effort was redirected to areas that could still benefit by studies related to environmental systems cooling. These studies included investigation of:

- Heat capacity of cryogenic droptanks
- Cryhocycle system comparison
- APU Systems comparison
- Environmental systems cooling techniques for use during reentry and ferry phases of the flights

1.3 SUMMARIES AND CONCLUSIONS

From the initial studies, the following conclusions were developed:

- Heat balance studies. A comparison of the rate and cumulative heat generated with the rate and cumulative cryogens usage showed that a basic incompatibility exists and that cryogens cannot be used to absorb the generated heat as they are required for use.

- Cryogens usage management. Several comparisons were made of concepts that used the cryogens in different ways and compared these concepts with a baseline system which employed radiators. The baseline system utilized dedicated vented hydrogen to provide cooling when the radiators were inoperable. One of the studied concepts utilized dedicated hydrogen in addition to normally vented hydrogen for cooling instead of radiators. This turned out to be 1200 lb heavier than the baseline but the system did not have the deployment and operational problems associated with the radiators.

  Another concept used accumulators to store the ACPS cryogens after they had been conditioned by the EC/LSS heat in conjunction with dedicated hydrogen for additional cooling. This system turned out to be about 8400 lb heavier than the baseline system.

  Other concepts which utilize larger accumulators were considered but they were extremely heavy.

  Optimization of combinations of low flowrate and high flowrate studies were started but not completed because of the change in Shuttle configurations. Approximations indicated that such a system would not be significantly lighter than the first one mentioned above.

- Heat capacity of ascent tanks and residuals. The analysis was oriented toward determining the practicability of using the tanks (1) to store environmental control and equipment waste heat and (2) to make this heat available at an appropriate rate for conditioning of ACPS propellants. The analysis showed that over 2 million Btu could be absorbed by the ascent tanks before a temperature of $500^{O}R$ would be reached and that in order to transfer heat from the tanks to the ACPS propellants at the high rates required

large heat exchangers and compressors woul be necessary. This system appeared cumbersome and little hope was felt that it would result in significant weight advantages.

From the studies that are applicable to the current Shuttle configurations, the following conclusions were reached:

- Cryogenic/Freon heat exchanger. Early in the study, an effort was initiated with the AiResearch Manufacturing Company to parametrically investigate several hydrogen/Freon and oxygen/Freon heat exchangers capable of transferring EC/LSS heat to the cryogenic fluids. Many of the parameters were selected on the basis of pressure and flowrates established by the Phase B Shuttle contractors. However, the parameters were broad enough to be applicable to current Shuttle design conditions. The study showed that cryogenic hydrogen/Freon and oxygen/Freon heat exchangers could be adequately designed, and significant development problems are not expected. The heat exchangers are compact and light.

- Radiators supplemented with refrigerator. A brief study was made to evaluate the extent to which a refrigerator could supplement the radiator for rejection of heat from the environmental thermal control system. The main idea is to increase the average radiator temperature by using an active refrigerator and thereby reducing the radiator area. The general conclusion is that a refrigerator will not sufficiently aid the system to warrant the added complexity; however, for configurations in which radiator area is a significant problem, there may be no other choice.

- Cryhocycle comparisons. Comparisons were made between a baseline system consisting basically of fuel cells for power and radiators for heat rejection and Cryhocycle system which uses a cryogenic hydrogen expander to provide both power and cooling. The resulting weight comparisons showed that the baseline system was lighter by about 442 to 534 lb, depending on the basic data. However, the

Grumman Corporation also performed a cryhocycle study with slightly different assumptions and showed the two systems to be approximately equal in weight.

- APU Comparisons. Three APU systems were compared on the basis of weight. Functions of both power generation and EC/LSS cooling during deorbit and reentry were considered. The three types of APUs and a cooling system which uses dedicated cryogenically stored hydrogen that is heated and vented overboard, a hybrid hydrogen APU that expands part of the hydrogen that is used for cooling, and a cryogenically stored oxygen and hydrogen-supplied APU that utilizes the EC/LSS and APU generated heat to condition the reactants. The study showed that the oxygen-hydrogen APU was the lightest by 770 lb as compared to the hydrazine APU system, and the hybrid system was 320 lb lighter than the hydrazine APU system.

- Ram Air Cooling. To better define how much dedicated fluid would be required during reentry, an investigation was made to (1) determine the capability of achieving rejection of the EC/LSS heat to air during descent by means of passing ram air between the folded and stowed radiators and (2) the possibility of cooling the hydraulic oil only by means of a fin-and-tube oil-to-air heat exchanger.

  For the first study, it was concluded the the ram air could not adequately be used to cool the stowed EC/LSS radiators. This was due to the large area requirements associated with ram air cooling of the Freon in the radiators, and the relatively low Freon-to-air temperature difference, high heat loads, and absence of fin convective effects, and the inability of achieving the desired Freon outlet temperature below about 13,000 ft.

The results of the second study indicated that ram air cooling over 17 in. by 17 in. by 3 in. thick heat exchanger could be used for APU cooling below 56,000 ft.

- EC/LSS - APU cooling during reentry. Several concepts to provide cooling for the EC/LSS and APU systems during deorbit and reentry were reviewed. The application of these concepts to other phases of flight, such as the horizontal ferry flights and flight tests, was also evaluated. Of the several systems that employ expendable fluids (hydrogen, water, ammonia, etc.) for cooling, the hydrogen system appears to be the best. It is light and requires a minimum of new technology for development and is applicable to all phases of flight. Various compressor-expansion machines were considered, and the one that appears best is a closed-cycle vapor compression refrigerator that uses water for cooling outside the atmosphere and air within the atmosphere.

1.4 OVERALL CONCLUSIONS

- Radiators cannot be efficiently replaced by cryogenic cooling techniques.

- Ascent tanks can be employed as heat sinks during ascent and the first hour or so of orbital operation, but the added complexity associated with heat exchanger and circulation systems does not justify the slight weight savings.

- A Cryhocycle system does not provide a major advantage, and considerable development would be required.

- The lightest and simplest cooling system for EC/LSS heat control during deorbit and reentry is one which utilizes hydrogen for the expandable fluid.

- APU cooling can also be accomplished by the use of the same expendable hydrogen. A minimum number of technology efforts would be required if this approach were taken.

- Consideration should be given to the use of a separate APU cooling system because of the higher heat rates and higher temperatures of that system as compared to the EC/LSS cooling system.

- Water and ram air provide the best coolant for a separate APU cooling system if cooling with hydrogen is ruled out.

1.5 REPORT ORGANIZATION

This report has been organized according to the various studies accomplished on the various subsystems and concepts. Each study is treated as an entity in itself with the level of detail being different for each. In general, most studies were conceptual. This was due to the fact that many different systems were studied and two different Shuttle configurations were in existence. In reporting these studies, they have been grouped according to their applicability to the early Phase B fully resuable Shuttle configurations (Section 2) or to the current configuration which consists of a resuable orbiter with droptanks (Section 3). Thus, in Section 2, studies can be found that pertain to internal cryogenic ascent tanks, and hydrogen- and oxygen-supplied OMPS, ACPS, and APU, whereas in Section 3, the studies pertain to cryogenic external ascent droptanks and smaller subsystems that employ cryogens. The OMPS, ACPS, and APU utilize earth-storable propellants rather than cryogenic hydrogen and oxygen.

Some of the studies initiated for the Phase B configurations were applicable to both Shuttle configurations (such as the cryogenic hydrogen/Freon heat exchanger studies and the Cryhocycle System Studies) in which case they are described in Section 3.

Section 2

STUDIES APPLICABLE TO PHASE B SHUTTLE CONFIGURATIONS

2.1 INTRODUCTION

At the time the Cryogenic Cooling in Environmental Control Systems (TASK 1A) was initiated, the Phase B Space Shuttle studies were in progress and nearing completion. The orbiter configurations that were used as a basis for the Shuttle Cryogenic Supply Systems Optimization Study (SCSS) were taken from the Phase B Studies and were also used for the early Task 1A studies.

2.2 PHASE B ORBITER CONFIGURATION DESCRIPTIONS

The orbiter configurations as modified for the SCSS studies are shown in Figure 2-1. Two orbiter versions were considered: the McDonnell-Douglas (MDC) version and the North American-Rockwell (NAR) version. These orbiters used cryogenically stored oxygen and hydrogen for all propulsion systems and for the fuel cell (FC) and auxiliary power unit (APU) reactants. The main engines, which were ignited after the fully reusable booster depleted its propellants, drew their propellants from ascent tanks inside the orbiter vehicle. These are the large LO_2 and LH_2 tanks shown in Figure 2-1. After orbit injection, the main engines and tanks normally were no longer used. The Orbit Maneuver Propulsion Systems (OMPS) were also supplied with cryogenically stored hydrogen and oxygen: these tanks are the smaller ones shown in the figure. The Attitude Control Propulsion System (ACPS) drew its propellants from the OMPS tanks. The ACPS thrusters were designed to use oxygen and hydrogen gases, and therefore the cryogenically-stored propellants were pressurized by pumps driven by oxygen and hydrogen turbines and heated by heat exchangers which used the hot oxygen and hydrogen combustion products from the turbine exhaust and/or separate gas generators. The conditioned gases were stored in accumulators, sized to be compatible with the various duty cycles.

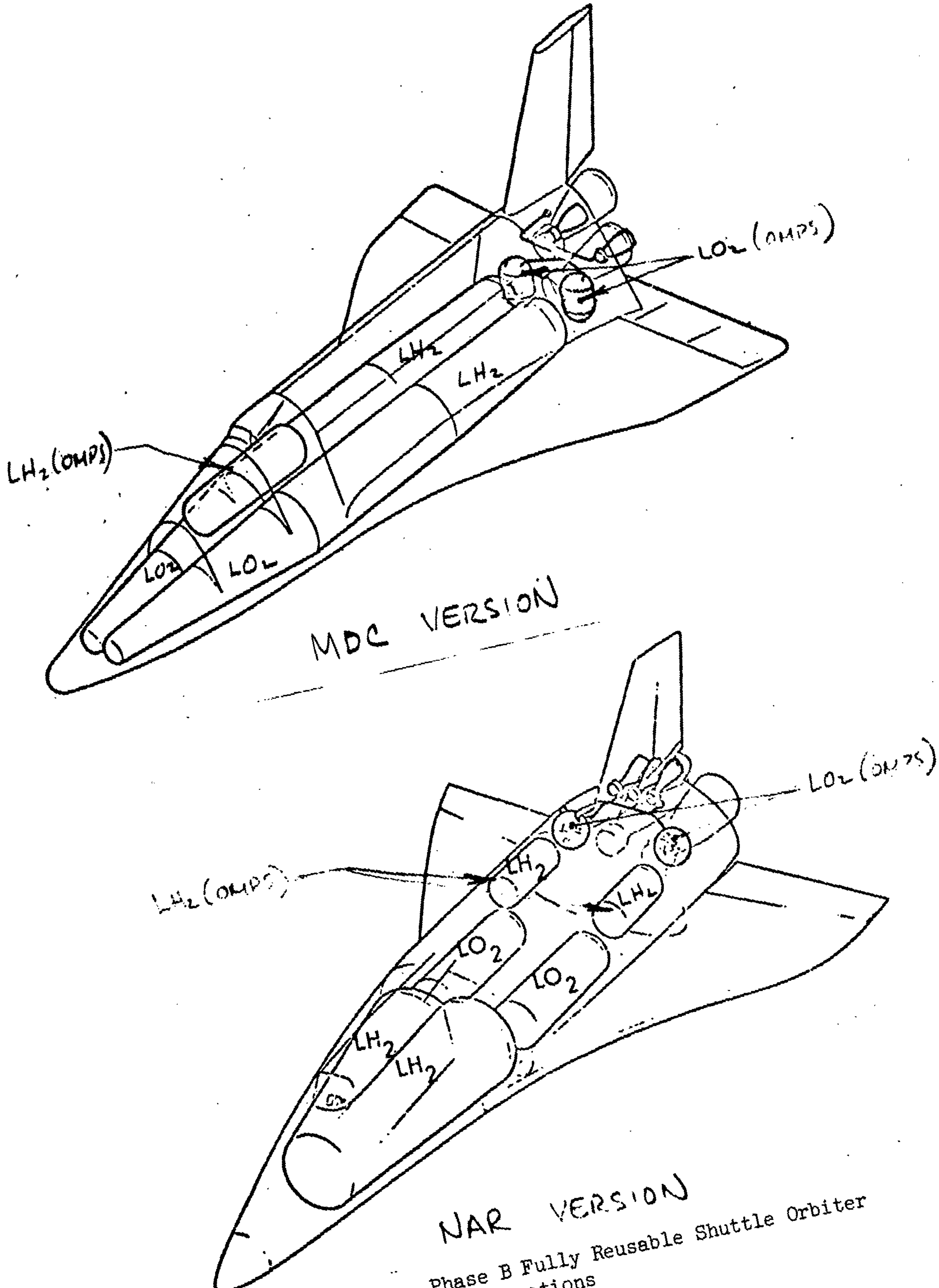

Fig. 2-1 Phase B Fully Reusable Shuttle Orbiter Configurations

The fuel cells were supplied with reactants from the accumulators, and the life support oxygen was supplied from the oxygen accumulator. Several options for the APU existed; the reactant could be drawn from the accumulators or from separate subcritical or supercritical tanks.

For the Phase B configurations, the Environmental Control and Life Support System (EC/LSS) heat rejection, which includes waste heat generated by the electronic system, was performed by space radiators; these were stowed within the payload bay compartment and required that the compartment doors be opened to deploy the radiators. In some configurations, the radiators were attached to the inside of the doors; in such cases, the doors had to remain open; other configurations deploy radiators mounted separately. In some cases the radiators caused undersirable operational restrictions.

Specific propellant quantities, vehicle criteria, and requirements are given in SCSS final report Volume II, Section 5. The assumptions and conditions pertinent to each study is contained in the discussions associated with each study.

2.3 STUDIES

2.3.1 Heat Balance Studies

One of the primary studies was to examine the relationship between the heat rate and total heat being generated on the orbiter and the capacity of the cryogens to use this heat. In the process of defining these relationships, a model of the EC/LSS heat rejection loop was assumed. The simplified Freon thermal control loop is shown in Fig. 2-2. Some typical heat rates are shown in the figure and the range of expected heat rates is shown in Table 2-1. The areas of study for the heat rejection loop are indicated in the figure.

It became evident that cryogenic heat exchangers were of major concern and a study of cryogenic/Freon heat exchanger was initiated with AiResearch, as discussed in Section 3.

A nominal heat load obtained from the NAR Phase B studies is shown in Table 2-2. It can be seen that the values lie between the ranges established in Table 2-1.

To develop the approach of how best to use the heat and cryogens, a cryogens-use schedule was constructed, as shown in Table 2-3. The use profile is based on a 7-day, 17th orbit rendezvous mission. The orbiter was assumed to dock with the space station, discharge its crew and/or cargo, and then separate from the space station. It would maintain a station-keeping mode in proximity to the space station for a major portion of the 7-day period (~ 127 hours) and then re-dock just prior to separation and the return-to-earth phase. Maximum and minimum quantities were taken from requirements and criteria sections of the SCSS study. The ACPS propellant defined therein was modified to represent only the propellant demanded by the thrusters. The total ACPS maximum propellant was based on an impulse of 1,687,000 lb-sec steady-state and 1,018,000 lb-sec pulsing at an average I_{sp} = 410 sec. The ACPS minimum propellant was a ratio from this base in accordance with the maximum and minimum established in the task reports. The fuel cell and APU consumption were changed slightly to reflect later Phase B work. The time steps shown reflect major phases rather than points where major cryogen consumption occurs; however, in some cases, the consumption and phase do coincide.

Using the cryogen and heat-generation profile, preliminary cumulative heat-balance profiles were constructed. To select a reasonable profile, a summary heat-balance chart was made and is shown in Table 2-4. This table shows (1) the rough order of heat generated by the fuel cell and electronics (assuming that all power generated by the fuel cell results in heat at the electronics); and (2) the heat capacity of the cryogens (assuming that they can be heated from near-saturated liquid conditions to a high-temperature gas). In this table, no regard was given to what could or could not be used, but rather what the heat capacity would be if all the cryogens could be heated. The heat capacities shown in the first two columns are based on the cryogen quantities noted in Table 2-3. For the boiloff and propulsion system cooling, the heat capacity shown is based on heat-rejection rates from the subsystems

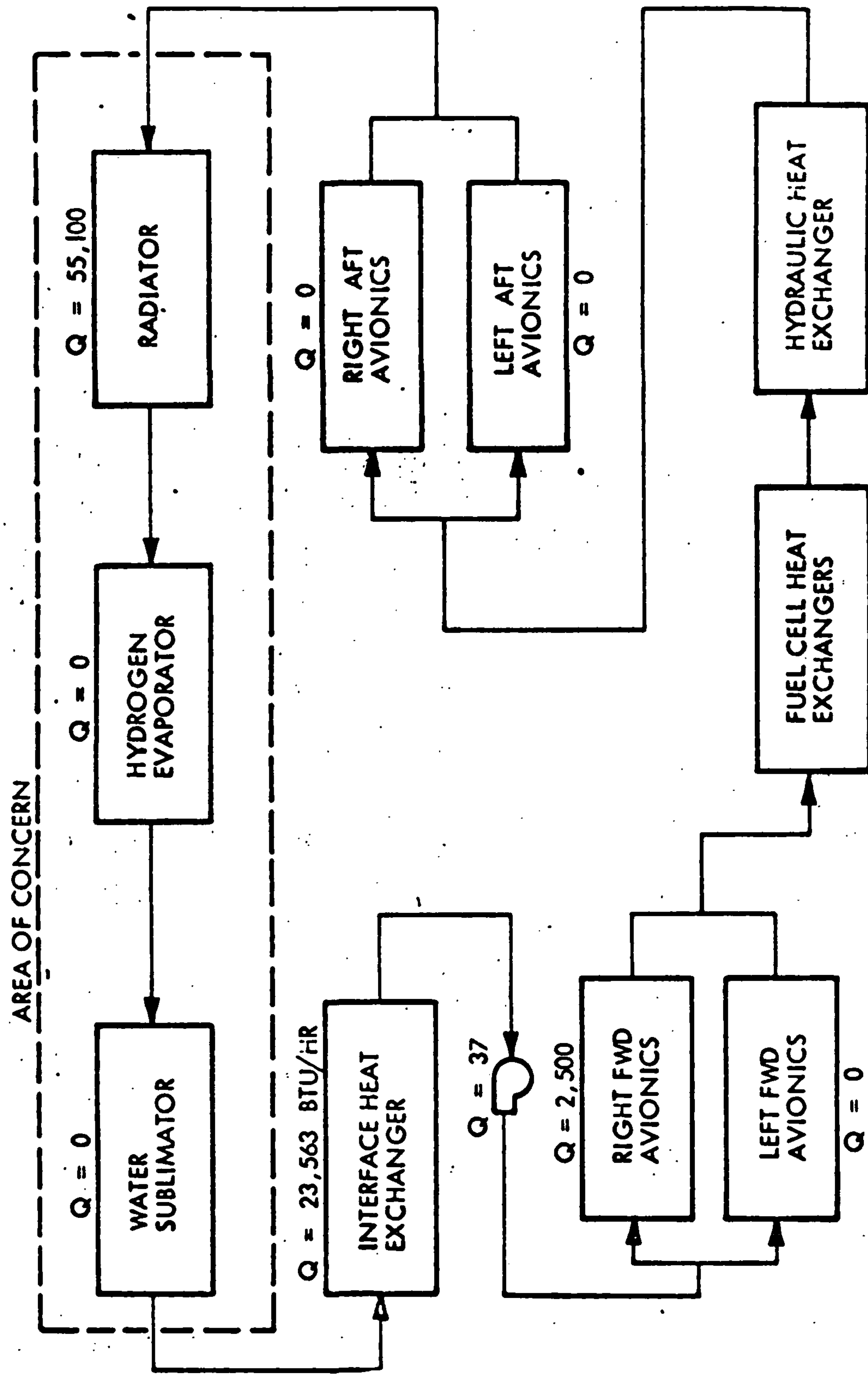

Fig. 2-2 Basic Thermal Control Loop

Table 2-1

Ranges of Thermal Control System Loads

MISSION PHASE	DURATION (HR)	CABIN HEAT LOADS (BTU/HR)				ELECTRONIC HEAT LOADS (BTU/HR)				TOTAL LOAD (BTU/HR)	
		CREW		WALLS		ELECTRONICS(3)		FUEL CELL WASTE(4)			
		MIN(1)	MAX(2)	MIN	MAX	MIN	MAX	MIN	MAX	MIN	MAX
PRELAUNCH	1.5	860	2600	0		24,200	34,130	14,880	24,000	39,940	62,730
ASCENT	0.12	860	2600	1000	2000	24,270	36,860	14,920	25,920	41,050	67,380
ORIENTATION PHASING	20.0	860	2600	-1000	1000	19,440	43,680	11,910	30,720	31,210	78,000
RENDEZVOUS DOCKING	3.0	860	2600	-1000	1000	26,140	43,350	16,100	30,480	42,100	77,430
ORBIT OPERATION	121.4	860	2600	-1000	1000	18,390	23,300	11,300	16,420	29,550	43,320
UNDOCKING/ PHASING	22.0	860	2600	-1000	1000	19,440	44,370	11,970	31,200	31,270	79,170
ENTRY (HCR)	1.5	860	2600	1000	2000	28,600	33,450	17,600	23,520	48,060	61,570
LANDING	0.75	860	2600	1000	2000	28,600	32,090	17,600	22,560	48,060	62,250

(1) BASED ON 2 MEN LOW-METABOLIC RATE
(2) BASED ON 4 MEN HIGH-METABOLIC RATE + 400 BTU/HR FOR LiOH REACTION
(3) ALL ELECTRIC ENERGY ASSUMED TO DISSIPATE AS HEAT
(4) FUEL CELL WASTE ESTIMATED AT 2,100 BTU/KW-HR (MIN) AND 2,400 BTU/KW-HR (MAX)

Table 2-2

Coolant Loads and Radiator Temperature

ITEM	MISSION TIME HOURS 0 1	1 22	22-26	26-29	29-159		159-164
Load on Humidity Control HX - BTU/Hr							
Electrical	283	283	283	283	283	283	283
Latent	980	980	980	980	490	490	980
Sensible	1,800	1,800	1,800	1,800	1,800	1,800	1,800
TOTAL	3,063	3,063	3,063	3,063	2,573	2,573	3,063
Load on Cabin HX Total - BTU/Hr							
Electrical	6,500	6,500	6,500	6,500	3,400	3,400	6,500
Non-electrical	2,000	2,000	2,000	2,000	1,310	1,310	2,000
TOTAL	8,500	8,500	8,500	8,500	4,710	4,710	8,500
Electronic Load on Water Loop - BTU/Hr	11,500	10,500	12,000	12,000	4,600	4,600	12,000
TOTAL LOAD ON WATER LOOP	23,063	22,063	23,563	23,563	11,883	11,883	23,063
Electrical Load on Freon Loop - BTU/Hr	2,900	2,500	2,500	2,500	2,040	2,040	2,800
Fuel Cell Electrical Load - Watts	9,600	9,200	10,600	9,000	4,100	4,100	7,500
G.E. Fuel Cell Heat Load - BTU/Hr	26,000	24,000	29,000	24,000	10,000	10,000	19,000
TOTAL RADIATOR LOAD WITH G.E. F/C - BTU/Hr	51,963	48,600	55,100	50,100	24,000	24,000	45,363
Avg. Radiator Temp. with G.E. F/C - °F	90	87	94	89	61	76	83
P&W Fuel Cell Heat Load - BTU/Hr	20,000	19,000	21,000	18,000	7,000	7,000	15,000
TOTAL RADIATOR LOAD WITH P&W FUEL CELL - BTU/Hr	46,000	43,500	47,000	44,000	21,000	21,000	41,400
Avg. Radiator Temp. with P&W F/C - °F	84	82	85	82	58	71	79

NOTE: WCP = 470 Except Where Noted.

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TABLE 2-3

MAJOR CRYOGEN CONSUMABLES

MISSION PHASE	TIME		DURATION	OMPS CONSUMPTION 15K AT 444 SEC		ACPS CONSUMPTION				FUEL CELL		APU	
						ΔV		MANEUVER					
	BEGIN	END		MIN	MAX	MIN	MAX	MIN	MAX	MIN	MAX	MIN	MAX
PRELAUNCH	1.0	0	1.0							1.7	2.4	45	105
ASCENT	0		:48							1.2/5.5	1.4/6.6	56	88
PHASING	0:48		21:24	3,620	4,340			35	77	171	228(2)		
HEIGHT	22:12		0:46	5,150	6,116			44	106	6.9	8.3		
COELLIPTIC	22:58		1:36	4,230	5,064			44	88	14.3	17.3		
TPI/MC	24:34		0:34	1,350	1,619	264	754	44	91	5.0	6		
REND /DOCK	25:08 26:36		1:28 2:09 3:37			88	215	44	91	13.6 16.4	16 20		
SEPARATION	28:45		0:15			88	215	35	55	2.0	2.4		
STATION KEEP	29:00		127:37					224	699	442	121(3)		
TPI/MC	156:37		0:55	1,130	1,354	264	1,278	44	91	8.0	9.6		
REND /DOCK	157:32 162:00		4:28 2:30			88	213	44	91	38.7 16.2	45 19.5	8	9
SEPARATION	164:30		1:10			88	213	35	55	7.2	8.6		
DEORBIT	165:40		0:20	7,490	8,960			101	101	2.3	2.8		
ENTRY	166:00		0:30					448	1,081	3.1	3.7	93	96
DESCENT	166:36		1:15							8.1	9.7	271 73	417 135
LANDING	167:45		0:15							2.0	2.4	60	117
GROUND CONTROL	168:00												
RESERVE				250	301	176	1,026					54.1	77
TOTAL				23,220	27,754	1,056	3,914	1,142	2,626	765	1,625	663	1,044

(1) ASSUMED 20% LARGER THAN MIN. EXCEPT AS NOTED.
(2) ASSUMED 12 KW
(3) ASSUMED 11 KW

TABLE 2-4

HEAT GENERATION AND HEAT CAPACITY

MILLION BTU (168 HR 17TH ORBIT RENDEZVOUS)

		MIN.		MAX.		INTEGRATED SYSTEM I		MDAC		MODEL USED	
HEAT GENERATION (FUEL CELL & ELECTRONICS)		4.83		10.72		4.83		4.04		4.83	
CONDITIONING TEMP.(1)		COOL	WARM	COOL	WARM	COOL	WARM	COOL	WARM	COOL	WARM
HEAT CAPACITY	ΔV	0.35	0.51	1.28	1.90	2.14	3.18	1.56	2.32	1.28	1.90
ACPS(2)	MAN	0.38	0.55	0.86	1.28			1.16	1.72	0.86	1.28
APU		0.44	0.68	0.70	1.07	0.41	0.63	0.66	1.02	0.44	0.68
FUEL CELL		0.18	0.26	0.38	0.55	0.38	0.55	0.15	0.22	0.18	0.26
BOILOFF		0.21		0.21		0.21		1.36		0.21	
PROPULSION SYS COOLING		0.80		0.80		0.80				0.80	
WATER											
RESIDUALS											
TOTAL		2.36	3.01	4.23	5.81	3.94	5.37	4.89	6.64	3.77	5.13

(1) COOL = H_2 AT 350° R; O_2 AT 390° R; WARM = H_2 AT 520° R, O_2 AT 520° R

(2) BASED ON AMOUNT DELIVERED AT THRUSTER

and integrated system studies. The boiloff of 161 lb was used, with approximately 0.8 of this being assumed as usable for additional heat absorption. The initial condition for cooling (after cooling propulsion system components) is at a low pressure and an approximate temperature and enthalpy of $40^{\circ}R$ and 100 Btu/lb, respectively. The gas was assumed to be heated to $520^{\circ}R$ with a $\Delta H = 1{,}650$ Btu/lb.

The columns "cool" and "warm" refer to how hot the gas was assumed to be and is identified in the footnote on the chart. The next two major columns contain data generated for the Integrated System Studies which were reported in the SCSS and by McDonnell-Douglas Phase B Space Shuttle reports. A reasonable model for preliminary analyses is shown in the last column. This model summarizes the time history balance of heat generated and heat-capacity plotted in Fig. 2-3. Data in this figure serve to illustrate the problem of the degree that the cryogens can be expected to be used as a heatsink for the generated heat. The fuel cell and electronics accumulated heat is shown as the top curve in the figure, and the cumulative heat capacity of the stored subsystem cryogens is shown as the crosshatched bank immediately below it.

The top line of the band is obtained by defining the ACPS reserves to be used at the first of the mission and the cryogens expended at the first part of each event. The bottom line of the band is obtained by assuming that the ACPS reserves are not used and that the consumed cryogens are at the end of each event. The band is composed of the sum of the heat capacities of the ACPS propellant, the APU reactant, the fuel cell reactant, and the hydrogen that must be vented because of propulsion system heat leaks. The heat capacity of all the cryogens, except the vented hydrogen, was based on heating them to $350^{\circ}R$ for the hydrogen and $380^{\circ}R$ for the oxygen. Heat capacity of the vented hydrogen was based on heating to $520^{\circ}R$ after it had performed its propulsion system cooling function. Temperatures of $350^{\circ}R$ and $380^{\circ}R$ for the heating capacity were based on preliminary heat-storage optimization analyses.

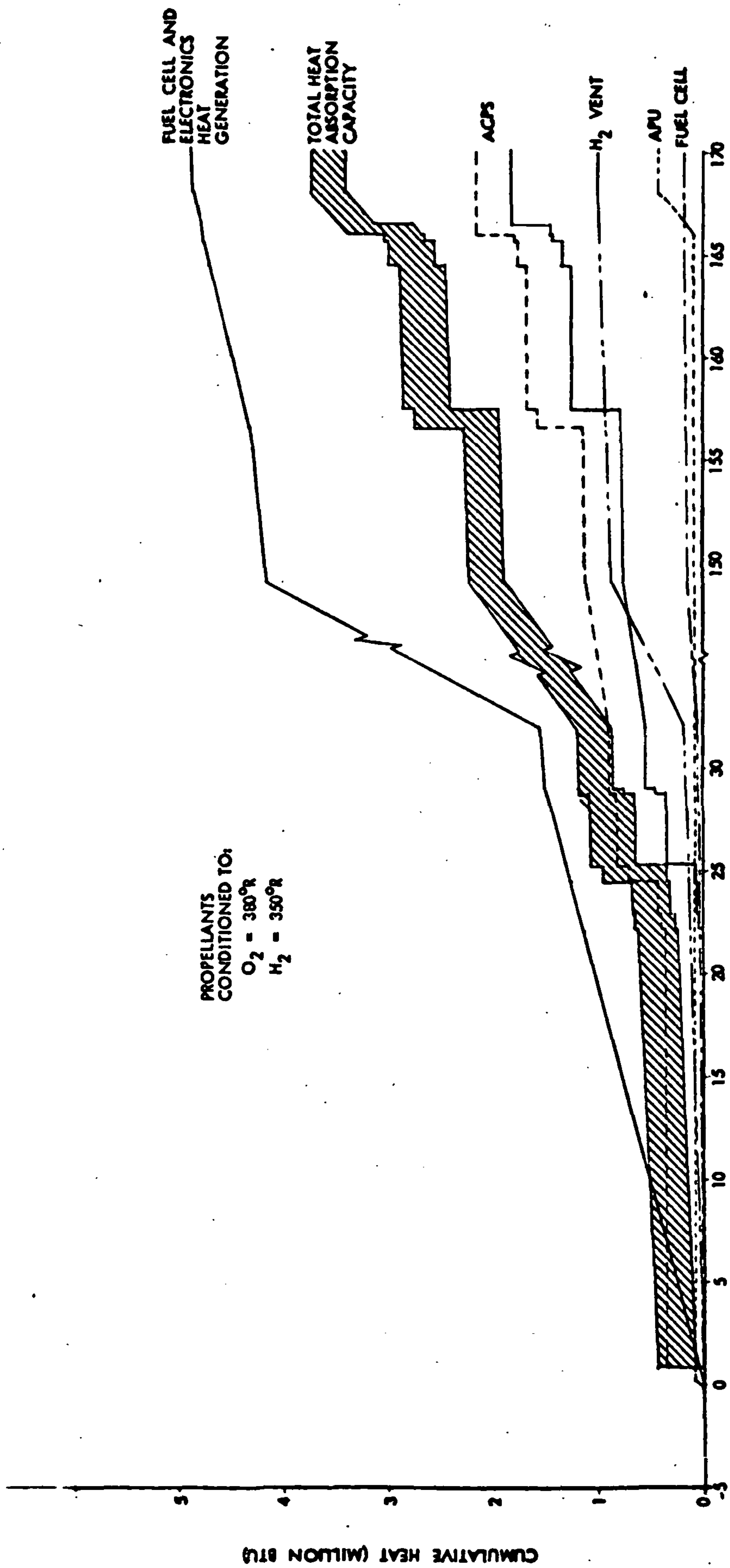


Fig. 2-3 Heat Generation and Absorption Capabilities

As can be seen from these data, there is a basic incompatibility between the rate that heat is generated and the quantities of cryogens used at each event that can absorb the heat. Two additional sources of cryogenics (not shown on the curve) that might conceivably be used as heat sinks are residuals in the ascent tanks and the OMPS propellants.

Analyses of the heat rate into the ascent tanks indicated that environmental heating for "warm" orbit conditions would cause the tanks to heat up and vent their residuals in 10 to 20 hours after lift-off or ground elapsed time (G.E.T.). For most performance analyses, large quantities of residuals and reserves must be maintained; however, the only remaining reliable fluids for thermal control are the true residuals, which consist mostly of pressurization gas and a small quantity of liquid in the lines. If EC/LSS heat is also added to the tanks, they will be at an average temperature of 450°R in about eight hours. The cumulative heat curve shown in Fig. 2-3 could be shifted to the zero mark at about the eight-hour G.E.T. point. This would better the relationship between heat generated and cryogens heat capacity as they are used, but still not resolve the incompatibility.

The OMPS propellants are used as liquid to either an RL-10 engine or a newly designed pump fed liquid fed engine. Therefore, no cooling could be accomplished by them.

Some ways to eliminate the incompatibility are: (1) have large isolated heat sinks on board the vehicle, which would be very heavy, (2) use large accumulators, which are also heavy, or (3) reduce the heat being generated, which can be done by the use of a Cryhocycle. These latter two approaches will be discussed in subsequent sections.

2.3.2 Cryogens Usage Management

Various concepts can be formulated that represent different degrees of integration for balancing the heat generated and the heat required for propellant conditioning. In Section 2.3.1 the heat generated and heat

requirements throughout the mission were compared. The cumulative heat generated and heat required curves were determined independent of each other; accordingly, care must be taken when integration concepts are formulated. As an example of this point, at the beginning of the mission the band representing the heat required is greater than the heat generated. Therefore, some preconditioned (or gas generator/heat exchanger conditioned) propellants must be stored to meet these demands. Since this represents about 400,000 Btu (maximum limit of band), this value of heat must be subtracted from the upper limit of the cumulative heat required band if integrated with the cooling system. For the minimum limit of the band, about 75,000 Btu must be supplied by preconditioned propellant, and this value must be subtracted from the lower limit of the cumulative heat required band if integrated with the cooling system. With these considerations in mind, weight estimates and descriptions for several concepts were made. These concepts are compared to a baseline concept.

2.3.2.1 Baseline Concept. The baseline concept provides that none of the heat generated is used to condition the propellants. Heat generated will be rejected from the vehicle by means of a coolant loop and space radiator. The propellant-conditioning requirements will be met by using high-flow rate pumps and gas-generator-supplied heat exchangers, with conditioned propellants stored in accumulators that supply the ACPS, APU, and fuel cells. Approximately 4.84 million Btu must be rejected by either the radiator or hydrogen for these concepts.

2.3.2.2 Integrated Concepts. The integrated concepts (i.e., concepts using all or part of the heat generated to condition propellants) are divided into two groups.

Group 1 concepts utilize large accumulators and low-flow pumps. The pumps are sized to flow propellant at a low rate, thereby allowing the propellant to absorb the heat (by means of a heat exchanger) at the same rate as the heat is generated. Since the propellants are not needed at the same rate as the

heat is generated, the accumulators are sized to accommodate the excess heat. No gas generators are required for heating the propellant.

Group 2 concepts utilize both high-flow and low-flow pumps. The low-flow pumps serve the same purpose as those for group 1 concepts and the high-flow pumps are used during peak propellant-requirement demand. Gas generators are used to heat the propellant during these peak demands. The accumulators are sized much smaller and do not have nearly as much heat-storage capacity as those for group 1 concepts. Excess heat generated is removed by means of dedicated hydrogen.

Group 1 Concepts. Two group 1 concepts were considered. The first concept contains accumulators sized to accommodate the propellant required for storing the excess heat generated. Preconditioned propellant (corresponding to a heat-storage capacity of 400,000 Btu) is loaded in the accumulators to meet the propellant requirements at the beginning of the mission, when propellant-conditioning heat requirements exceed the heat generated. The size of the accumulator is determined by the heat-storage requirement, which occurs at the end of the station-keeping periods and amounts to about 2,420,000 Btu. This concept requires that additional (and unused) propellant must be loaded to store the excess accumulated heat generated at the end of the mission (approximately 1.55 million Btu).

The second group 1 concept contains accumulators sized so that at the end of the mission the accumulators are empty, the excess heat generated having been removed by dedicated hydrogen vented overboard. The accumulator contains a heat-storage capability of about 850,000 Btu. Similar to the first concept, the accumulator will be loaded with preconditioned propellant (corresponding to a heat-storage capability of 400,000 Btu) to meet the propellant requirements at the beginning of the mission. This accumulator is sized so that prior to stationkeeping all heat generated can be stored in the accumulator. During stationkeeping, the heat generated will be greater than the accumulator storage capability, and dedicated hydrogen will be used to remove the excess

heat generated. During the stationkeeping, a total of about 1.55 million Btu will be removed by this hydrogen system. After the stationkeeping mode, no additional dedicated hydrogen is needed.

Accumulators required to hold the propellant associated with the heat-storage requirements quoted above for these concepts are excessively large. A rough estimate of the system weights were made for comparative purposes only, and no further analyses are envisioned for these concepts. Considering heat loads of 100,000 Btu and 500,000 Btu, the weights of hydrogen and oxygen required to absorb the heat, and of corresponding accumulators and accumulator residuals, were determined for propellant mixture ratios of 3.0 and 4.0. Initial H_2 and O_2 conditions were taken as $50^{\circ}R$ and $170^{\circ}R$, respectively, at 2,000 psia. The heat exchanger outlet pressure (accumulator storage pressure) was assumed to be 2,000 psia, and a range of outlet temperatures from $250^{\circ}R$ to $520^{\circ}R$ was investigated.

Results of the investigation are shown in Fig. 2-4. Mixture ratio apparently does not have a strong effect on the summed weight in the range studies, and minimum weight is seen to occur in the area of $350^{\circ}R$, which is the heat exchanger exit temperature. Comparison of the 500,000 Btu case with the 100,000 Btu case shows that the weights vary linearly with respect to the heat load, as would be expected.

Group 2 Concepts. As previously mentioned, the group 2 concept is a system wherein low-flow pumps circulate the propellants through heat exchangers and use vehicle waste heat for propellant conditioning. These pumps operate when accumulator capacity is available and Freon cooling is required. They would raise accumulator pressure from approximately 650 to 2000 psia. When the low-flow pumps are not operating, because the accumulators are being completely charged, a dedicated heat exchanger, using vented hydrogen, is used to condition the Freon.

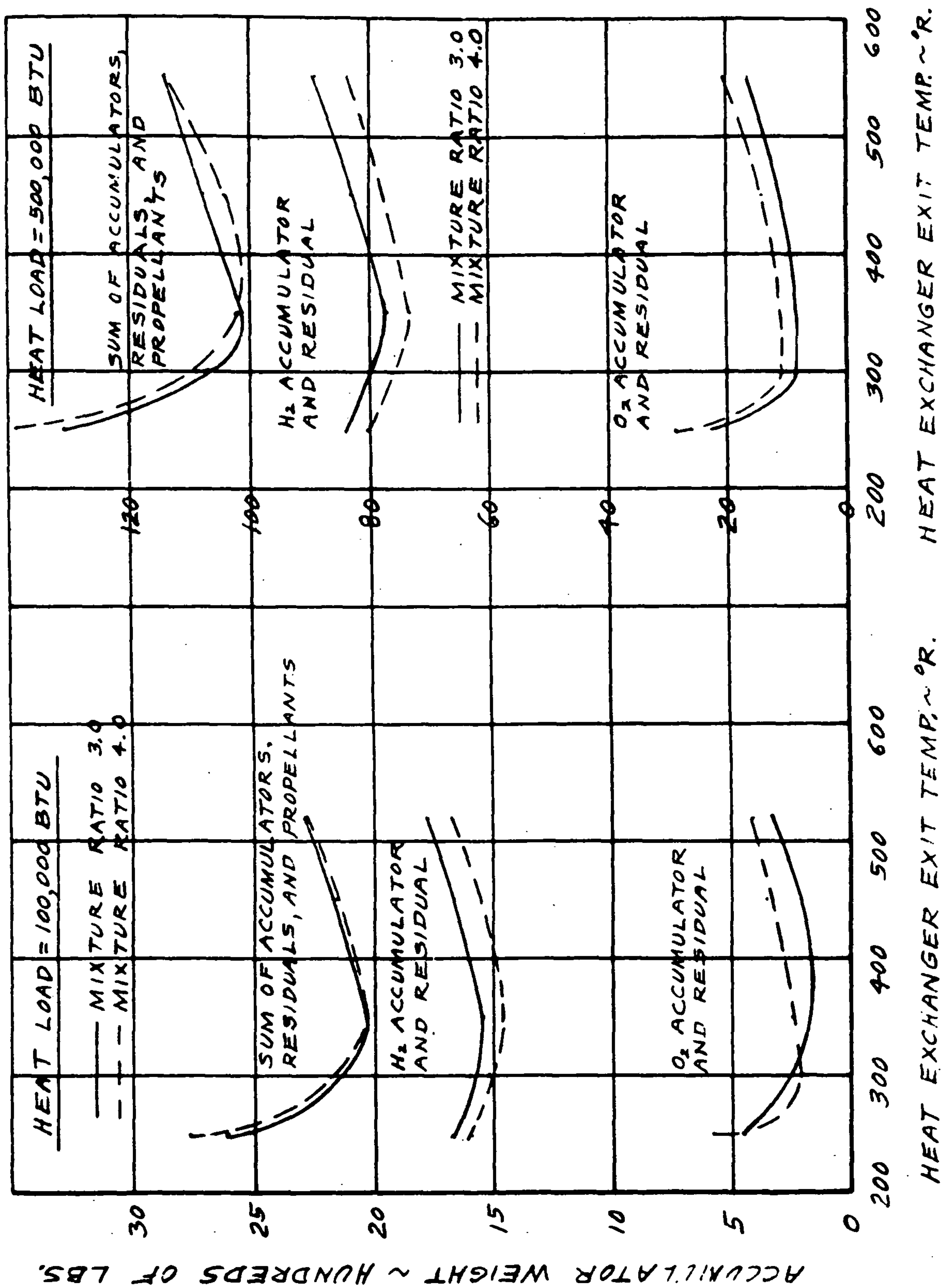


Fig. 2-4 Accumulator Weight to Store Heat

When ACPS high-flow demands are required, the normal ACPS turbopumps operate to supply propellants. For preliminary purposes, a nominal steady-state operating pressure of the turbopumps has been selected as 500 psia, and the pumps shutdown when 650 psia has been reached.

A baseline configuration has been established for the above approach (high and low flow pumps), see Fig. 2-5. This concept uses a Freon loop to absorb heat from the environmental control, life support, and fuel cell systems. The Freon is then circulated to heat exchangers for cooling - first by the vented hydrogen (HX 13), which is used to cool the tanks and ACPS pumps. The Freon is then circulated to the low-flow pump oxygen heat exchanger (HX 14), where it is used to condition oxidizer and then to the low-flow pump hydrogen heat exchanger (HX 12), where it is used to condition hydrogen. If additional cooling of the Freon is required, it is accomplished by venting hydrogen overboard through heat exchanger HX 11.

Preliminary analyses were conducted and resulted in the general arrangement of heat exchangers. Freon cooling takes place first in heat exchanger HX 13, because hydrogen will always be vented overboard for cooling and capacity remains in this gas, as it would have a probable maximum temperature of 150°R. Conditioning of the oxidizer was selected first on the basis of preliminary analyses, which indicated that per pound of accumulator dry weight, approximately twice as much Btu could be stored in an oxygen accumulator as compared to a hydrogen accumulator. Detailed analyses were not conducted to determine the proper sizing of accumulators, including the size ratios between the two propellants, the pressure and temperature to which the trickle pump should condition the propellants, and whether the preliminary selection of conditioning oxidizer and then hydrogen was correct.

The low-flow pumps should be optimized for size, taking into consideration probable duty cycles of the ACPS, the effect of off-time, and the interrelationship of power required to drive the pumps, and heat generated by the fuel cells in supplying the power. Preliminary analyses of these pumps indicate that they will have to be positive-displacement-type pumps because

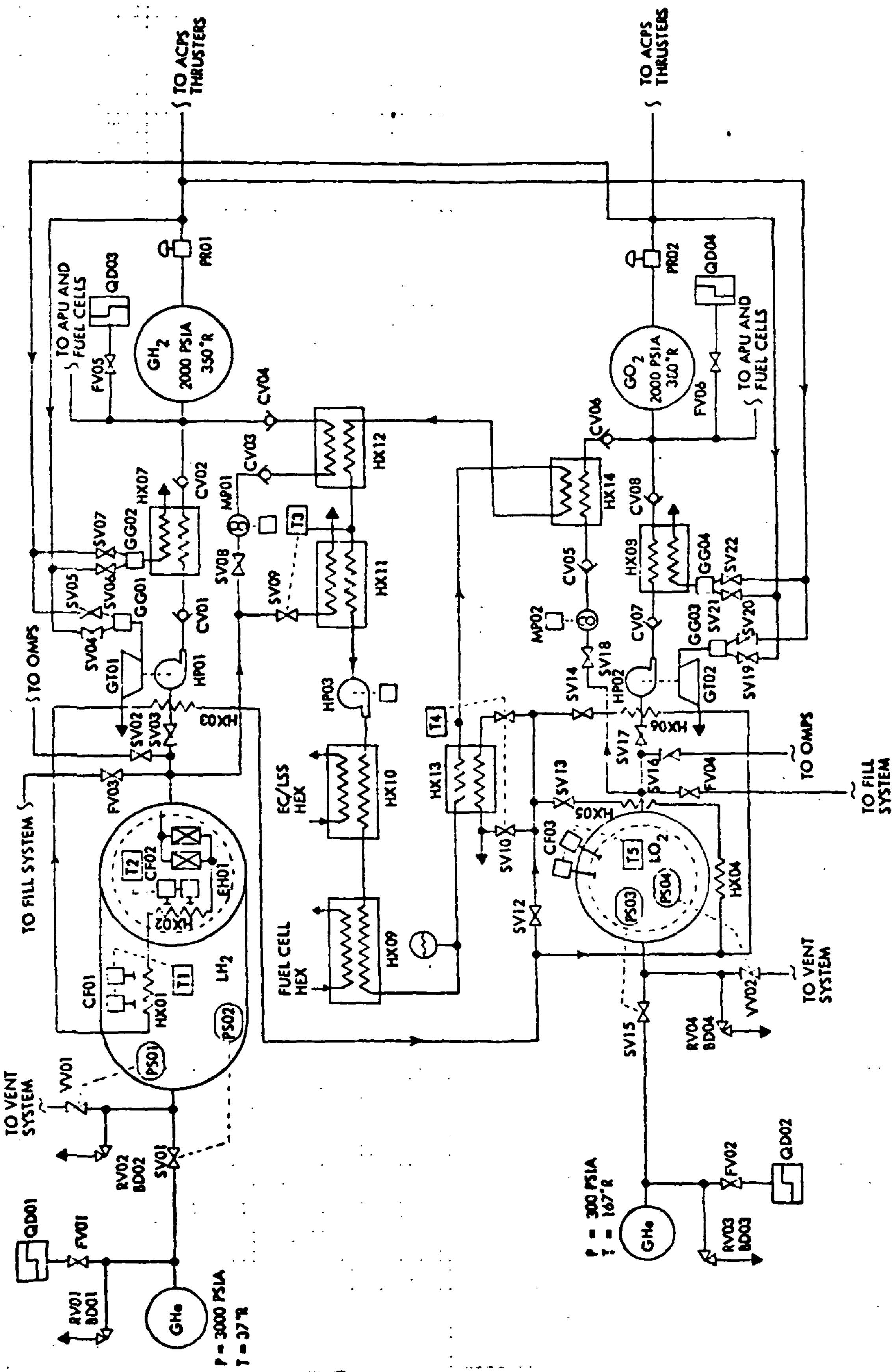


Fig. 2-5 Integrated Environmental Thermal Control System

of the low-flow rates and high head. They will probably have to be variable speed pumps with total pump power in the 1-to-2 hp range. Studies will consider relationships between pump power, flow rate, head rise, heat exchangers, and accumulator size.

Weight estimates were made of the baseline concept, a concept which utilizes dedicated hydrogen for cooling, and a concept that provides large accumulators to store waste heat conditioned propellants. The results are summarized in Fig. 2-6.

Case (1). In effect, this is a baseline approach and utilizes a space radiator on-orbit and vented hydrogen for the periods when the radiator is inoperable, i.e., first few hours of flight, after the radiator is being deployed and put on stream, and the last 2 hours, when the radiator is shutdown and stowed for re-entry. System weights would equal:

o Radiator	900 lb
o Vented LH_2	131
o Tank - ΔW	10
o Heat Exchanger	9
o Valves, etc.	43
	1,093 lb

Case (2). This approach assumed that no propellants are conditioned by ECS waste heat, but the Freon-21 circulating in the ECS/FC loop is conditioned (cooled) by normally vented hydrogen and any excess heat is absorbed by dedicated liquid hydrogen, which is vented overboard through a heat exchanger.

Results of past studies on this contract for the OMPS and ACPS showed that 504 and 183 lb of vented hydrogen are required to cool the ACPS turbopumps and the LH_2 and LO_3 propellant tanks, respectively. Since this gaseous hydrogen is at relatively low temperatures, and for Freon-21 cooling could be heated to approximately $500^{\circ}R$, it has a cooling capacity of approximately 1,000,000 Btu.

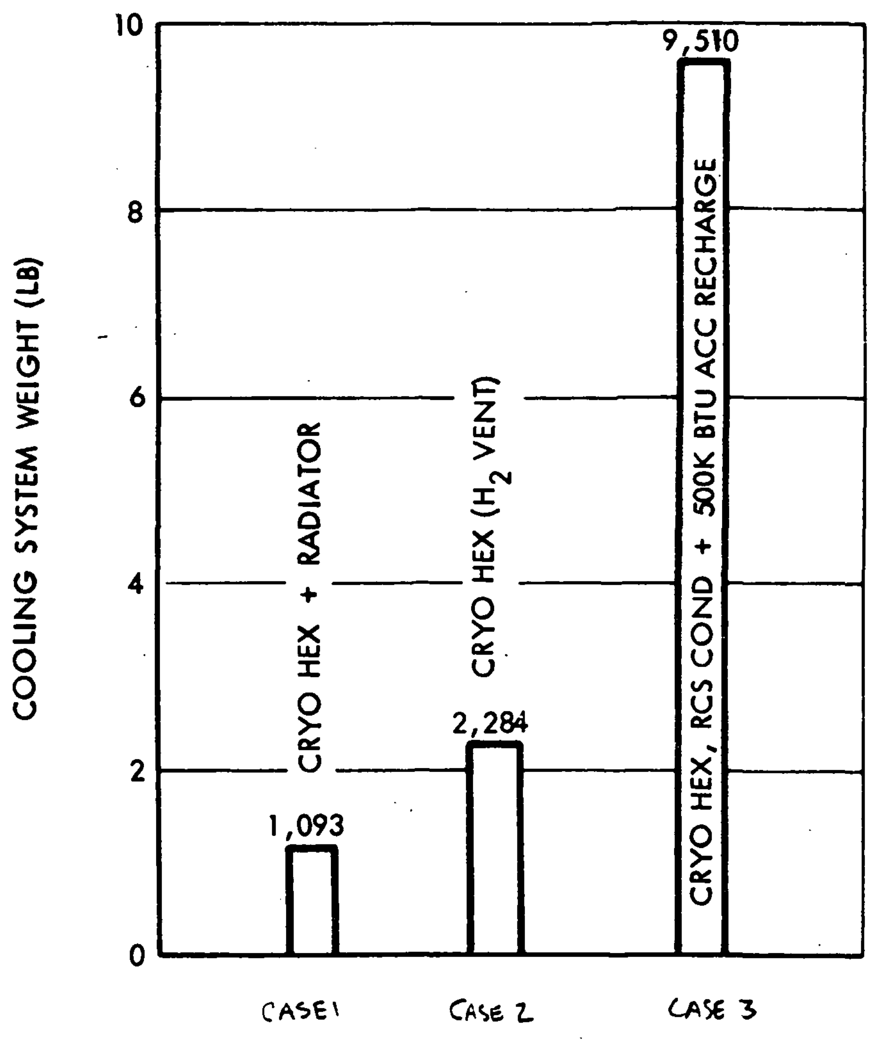


Fig. 2-6 Cooling/Propellant Conditioning Systems Weight Comparison

However, a total heat output for the Freon-21 cooling loop is approximately 4,800,000 Btu; there, 2,079 lb of hydrogen (based on heating to 500°R) would have to be vented overboard to absorb the excess heat. This dedicated cooling hydrogen would be stored in the main on-orbit LH_2 tank and would result in a slight increase in tank size. The total weight summary of the above approach is as follows:

o Dedicated LH_2	2,079
o Propellant Tank - ΔW	153
o Heat Exchanger Weight	9
o Valves, etc.	43
	2,284 lb

Case (3). This approach uses accumulators to store waste heat and small pumps and heat exchanger to recharge the accumulator to 2,000 psia. The oxygen and hydrogen were conditioned to 380 and 350°R, respectively.

The initial approach was to evaluate accumulators, which would store heat over a range from 500,000 Btu to the maximum that would accrue during any given period. The maximum was calculated to be approximately 2,400,000 Btu and would result in accumulators that would weigh approximately 53,000 lb if only hydrogen was conditioned, and the amount of hydrogen that would be stored would be approximately 1,300 lb at the entire ACPS nominal maximum requirement. Conditioning of oxygen only could reduce accumulator weight to approximately 19,000 lb, but this would result in storage of approximately 26,000 lb of oxygen, or a quantity that is approximately equal to the entire impulse oxygen needed for both the OMPS and the OMPS and the ACPS. Since this approach was deemed unrealistic, it was abandoned early, and an analysis was conducted on accumulators that would store 500,000 Btu.

The 500,000 Btu was stored in accumulators sized for the nominal 4:1 mixture ratio of the ACPS thrusters and was pumped to a maximum pressure of 2,000 psia. When the accumulators were fully recharged, dedicated hydrogen was vented overboard to absorb the Freon-21 cooling-loop heat, and at all times,

the normally vented hydrogen (pump and tank cooling) was used to absorb its maximum capacity of heat.

The above approach combined with the duty cycle, shown in Fig. 2-3 resulted in the requirement that 1,038 lb of hydrogen be vented overboard to cool during periods when the accumulators are full. The trickle pumps were sized to handle flowrates for the oxygen and hydrogen that could absorb heat rejection of 51,900 Btu/hr and resulted in sizes of 0.32 and 1.28 hp, respectively, for oxygen and hydrogen. These power requirements were based on an assumed efficiency of 80 percent and because of the low flow rates and high head requirements, positive displacement pumps are indicated. No technology assessment has been made, thus far, on pumps meeting these stringent requirements.

An interesting result came out of the above studies. Based on the assumed duty cycle and the accumulator size, minimal ACPS turbopump operation was required and indications are that capacity in the order to 750,000 Btu might eliminate ACPS turbopump operation for the selected duty cycle. The impact of this has not been assessed at present and is not to be construed as a recommendation that ACPS turbopumps could be eliminated. Elimination of the ACPS turbopumps could result in duty cycle limitations, and the stored heat approach does result in an extremely heavy system. The weight summary for the stored-head approach is as follows:

• Accumulators	9,241 lb
• Vented Hydrogen	1,038
• Trickly Pump	33
• Propellant Tank - ΔW	76
• Fuel Cell Reactants	172
• Heat Exchangers	13
• Valves, etc.	50
	10,623
ACPS Conditioning Propellants	-1,115
	9,508 lb

The above studies have shown that absorption of heat from the Freon-21 cooling loop by dedicated hydrogen or stored heat results in systems that are heavier than the baseline radiator system. However, these results are preliminary and analyses have not evaluated combinations of stored heat and vented hydrogen.

Additional studies in this area were not continued because of the Space Shuttle Vehicle configuration change.

2.3.3. Evaluation of Ascent Tank Heat Storage for EC Cooling and Propellant Conditioning

The potential and thermal practicability of using the orbiter ascent propellant tanks as orbit heat-storage tanks has been the subject of preliminary analyses and consideration. Analysis was based on the MDC APS Study ascent tank configuration and on the cumulative mission heat rates, as shown on Fig. 2-3. Three areas of investigation were pursued:

a. Heat sink capability of the tanks to meet the maximum cumulative excess heat condition of approximately 2.04 million Btu, occurring at approximately 156.5 mission hours.

b. Ability of the tank external heat exchanger configuration investigated by MDC to meet the propellant heating requirements to supply four 1,600 lb ACPS thrusters.

c. Relative heat-transfer capability of internal tank-wall forced convection versus the MDC external integral heat exchanger.

2.3.3.1 <u>Ascent Tank Heat Sink Capability</u>. The excess of available heat over required heat during the mission reaches a maximum of approximately 2.04 million Btu at 156.5 hours. Assuming $500^{o}R$ as a practical sink temperature limit, it was calculated that the heatsink would need to include both H_2 and O_2 tanks and all residual propellants. Fig. 2-7 shows the weight of residual H_2 and O_2, versus the temperature of the residual. Assuming an initial residual gas temperature of $200^{o}R$, the quantity vented during heating is

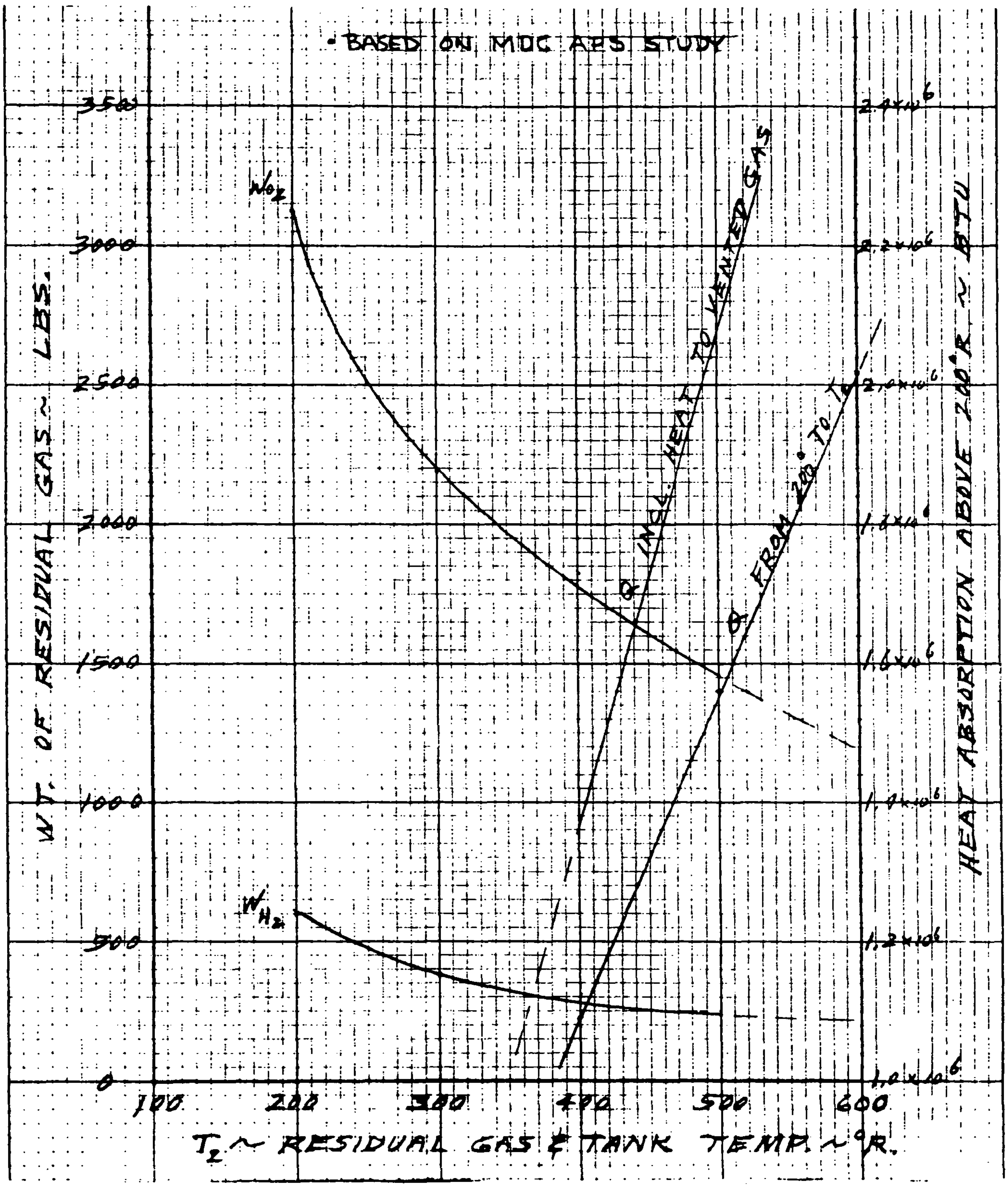


Fig. 2-7 **Residuals and Heat Capacity of Ascent Tanks**

obtained by subtracting the weight of residual at any value of T_2 from the weight of residual at $T_2 = 200^{\circ}R$. Also shown is the heat-absorption capability (Q) of the tanks and residuals versus T_2, assuming the heat input to start is $200^{\circ}R$. One curve of (Q) neglects the heat-absorption capability of the vented residual; the other Q-curve (the higher curve) takes credit for the heat carried away with the vented residual. When this higher, more realistic Q-curve is consulted, it is indicated that the system can absorb 2.04 million Btu without exceeding $500^{\circ}R$ sink temperature.

2.3.3.2 Adequacy of Tank External Heat Exchanger. Figures derived from MDC Low Pressure Auxiliary Propulsion Systems Study No. E0301 and MDC Preliminary Baseline Design Review, dated 17 December 1970, indicate that the maximum heat-exchange rates of the external heat exchanger are as follows:

$$\dot{q}_{H_2} = 1{,}430 \text{ Btu/sec}$$

$$\dot{q}_{O_2} = 790 \text{ Btu/sec}$$

The maximum heat-transfer rate into the tanks, taken from the slopes of the curves in Fig. 2-3 is approximately 15 Btu/sec, so that the storage rate would be adequately met by the external heat exchanger. The maximum rate of heat extraction from the tanks was calculated on the basis of four 1,600-lb engines operating at $I_{SP} = 410$ sec and a mixture ratio of 4.0, with required H_2 heating from $50^{\circ}R$ to $350^{\circ}R$ and required O_2 heating from $170^{\circ}R$ to $380^{\circ}R$. For this case of heat extraction, the desired heat-transfer rates would be:

$$\dot{q}_{H_2} = 3{,}520 \text{ Btu/sec}$$

$$\dot{q}_{O_2} = 1{,}275 \text{ Btu/sec}$$

Comparing these desired values with the available values derived from the MDC reports indicates that the external heat exchanger is inadequate to meet the desired heat extraction rates.

2.3.3.3 <u>Heat Transfer by Tank Internal Wall Convection</u>. The concept of a closed cooling system, using residual propellants to transfer heat between the internal tank walls and external non-integral heat exchangers, was envisioned as being potentially more effective than the MDC external integral heat exchanger. Preliminary calculations were made to evaluate this possibility. The MDC configuration utilized the exterior surface of the internally-mounted ascent tanks as a heat exchanger. Upon demand, fluid was passed through tubes, which were an integral part of the tank.

The effective thermal resistance in the MDC external integral heat exchanger, between the fluid in the tubes and the tank wall, was estimated from the thermal data provided in the MDC reports cited previously. The apparent resistances in this system are:

H_2 circuit: $R = 0.203$ sec-^{o}R/Btu

O_2 circuit: $R = 0.668$ sec-^{o}R/Btu

Against these values was compared the fluid film resistance that would occur between a moving stream of propellant along the internal surface of the tank wall and the wall itself. In this concept, the residual propellant flows through a closed-loop heat exchange system, transferring heat between the internal tank surfaces and a separate external heat exchanger. Hoop manifolds may be necessary inside the tanks to assure appropriate velocities longitudinally along the internal tank surfaces, and one or more external vapor compressors would be required. Using the MDC tank configurations for the orbiter, and assuming no heat transfer on the end domes and only 75 percent effective heat-transfer surface on the cylindrical H_2 tank section and the conical O_2 tank section, forced convection film coefficients were determined as a function of fluid velocity along the wall. Fig. 2-8 shows the corresponding wall gas film heat-transfer resistance as a function of gas velocity for H_2 gas in the H_2 tanks and O_2 gas in the O_2 tanks. Also shown in the figure are the apparent resistances of the MDC external heat exchangers. It is noteworthy that appreciably better heat-transfer rates appear to be available by

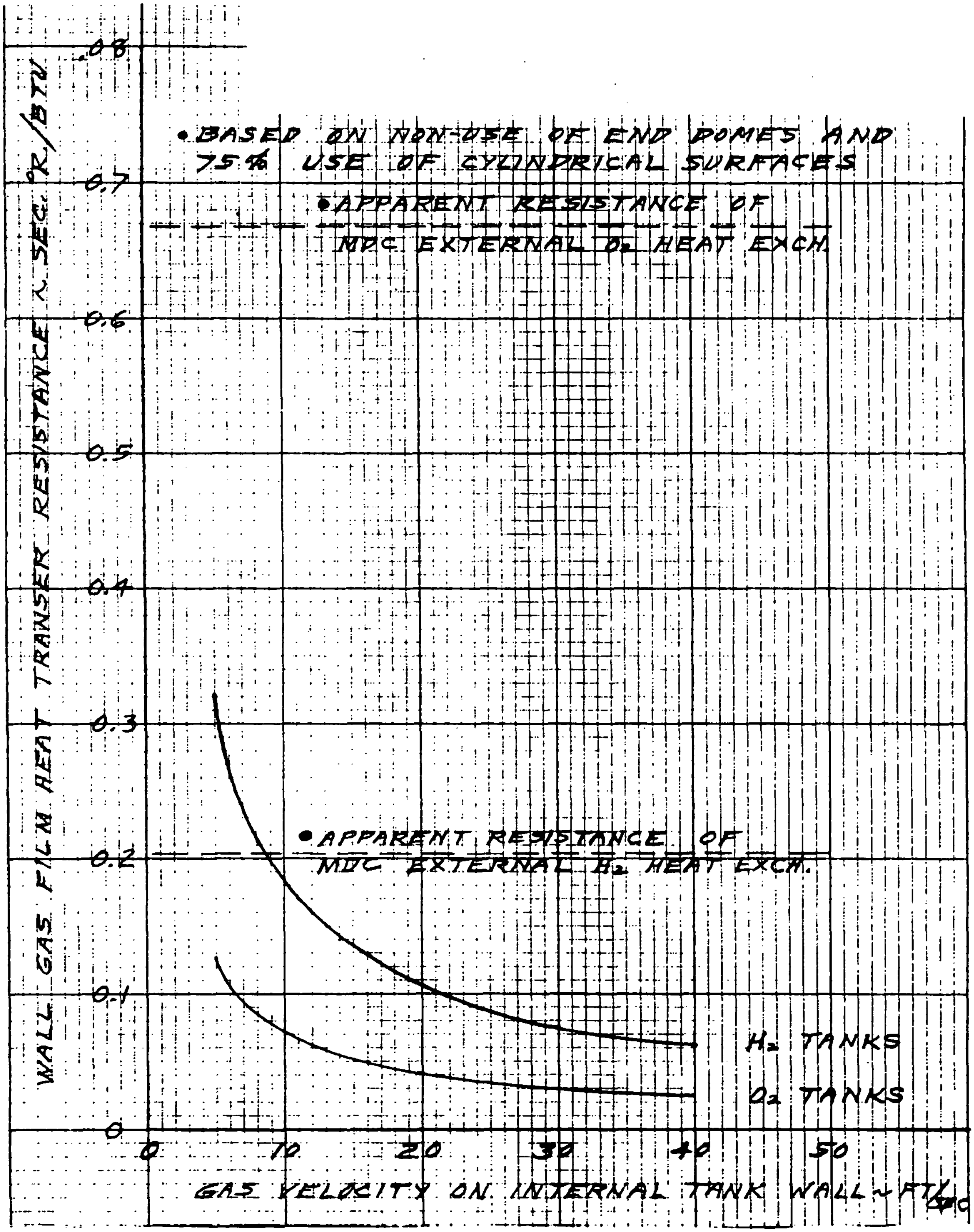


Fig. 2-8 **Thermal Resistance of Tank Internal Wall Heat Transfer Film Vs Film Velocity**

internal convective heat exchange, at modest velocities. Specifically, it appears that the maximum anticipated H_2 heat-transfer rate of 3,520 Btu/sec may be attainable at a velocity in the area of 25 ft per sec, and very low-velocity would be required to satisfy the maximum O_2 heat-exchange rate.

On the basis of the above, the internal convection scheme appears feasible and worthy of further analysis. The resistance of the external heat exchanger must be estimated and taken into consideration to obtain an overall system resistance, and a general refinement of the analytical techniques and system details must be incorporated.

Specific analyses were performed to determine the following:

a. Time from launch for the tankage to reach its maximum practical temperature for the intended purpose, considering both waste heat input and structural heat input.

b. Size of heat exchanger system required to transfer heat from the tankage to the ACPS propellants for conditioning purposes.

c. Size of heat exchanger system required to transfer heat from the environmental control and equipment cooling system into the tankage.

2.3.3.4 Data and Assumptions. This analysis was based on the use of McDonnell-Douglas Orbiter ascent-tank configurations, as they existed on 29 January 1971. The system for transferring heat into and out of the tankage consisted of closed-loop H_2 and O_2 fluid systems, respectively, transferring heat between external heat exchangers and the tank internal surfaces, as shown in Fig. 2-9.

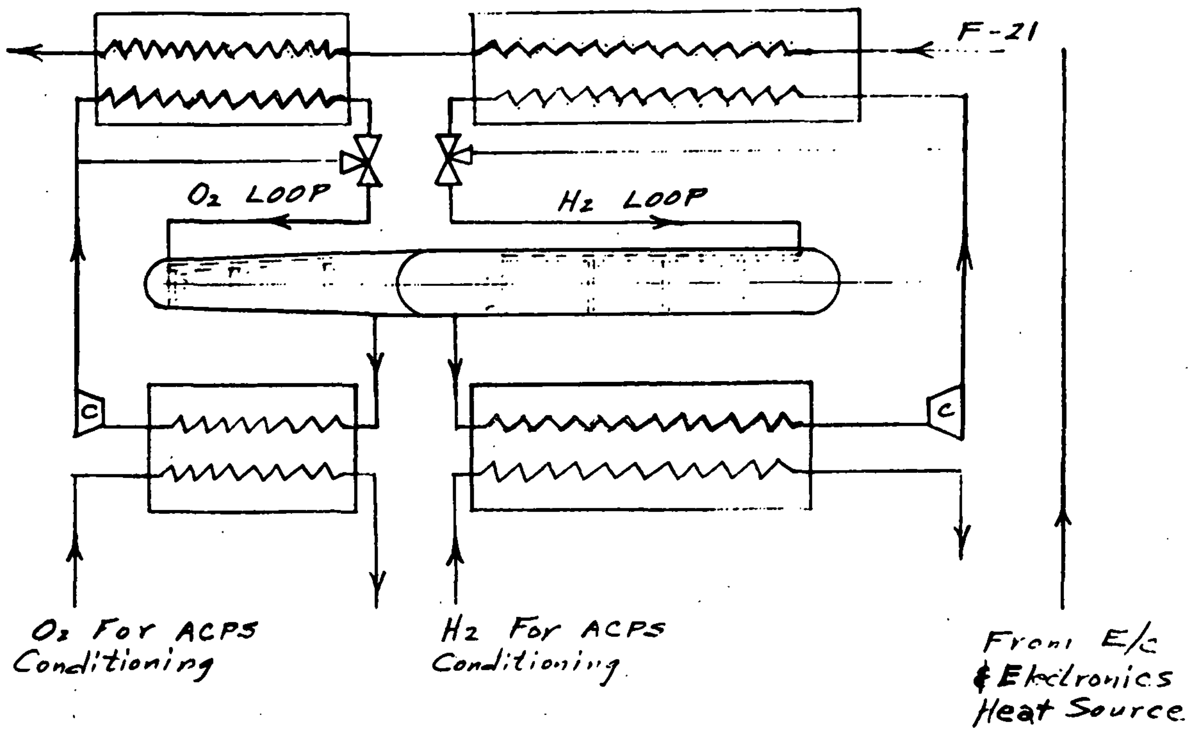


Fig. 2-9 Heat Exchanger Schematic for Heat Transfer to Ascent Tanks

Residual ascent propellants in the tanks were used as the circulating fluids. Assumptions were made as follows:

- Initial tankage and residuals temperature of 200°R
- All ascent tank external surfaces covered with 1-in. thickness of polyurethane foam having a density of 2.2 lb/ft^3
- Surrounding structure temperature of 500°R
- Maximum tankage temperature for heat sink purposes of 450°R
- Maximum environmental control and equipment heat-input rate to tankage of 54,000 Btu/hr, transmitted by the cooling of 0.5 lb/sec of Freon-21 from 610°R to 500°R in the tankage external heat exchangers
- Maximum heat extraction rate from tankage of 3,520 Btu/sec to condition 3.1 lb/sec of ACPS H_2 propellant from 50°R to 350°R at 2,000 psi, plus 1,275 Btu/sec to condition 12.5 lbs/sec of ACPS O_2 propellant from 170°R to 380°R at 2,000 psi. These propellant requirements are based on the operation of four 1,600 lb thrusters having an I_{sp} of 410 sec, with a mixture ration of 4.0.

2.3.3.5 System Description. The system concept investigated is shown in Fig. 2-9. Separate heat exchange loops are used for the H_2 and O_2 tanks. In each loop, the residual fluid is pumped by an external compressor through a conditioning heat exchanger, then through, or around, an ECS and electronics heat exchanger, and back to the tank. In each tank, the fluid is sprayed on the tank wall to impart or extract heat, as required. Heat is continuously imparted to the tanks by the ECS and electronics cooling fluid, at a rate of 54,000 Btu/hr. Heat for the ACPS system H_2 and O_2 propellant conditioning is extracted when required for ACPS operation. The analysis is directed at the applicability and the effectiveness of this system.

2.3.3.6 Tankage Temperature Rise. The assumption of a maximum tankage temperature of $450^{\circ}R$ for heat sink purposes was predicated on required Freon-21 outlet temperature of $500^{\circ}R$ from the ECS/electronics heat exchanger. Therefore, a practical limit exists for the tankage as a heat sink when a temperature of $450^{\circ}R$, bulk mean, has been reached.

For purposes of estimating tankage-temperature rise from time of termination of engine burn, the effects of structural radiation to the tanks, ECS/electronics heat input,and the sum of heat extraction requirements for all purposes were considered. The tanks were considered to be covered with one inch of polyurethane foam and an external sealer and to have a radiation emissivity factor of 0.5, which was estimated for the radiation coupling with the surrounding structure. Also, effects of tankage heat capacity and residual quantity versus temperature were considered.

Figure 2-10 shows tankage system heat capacities versus temperature, and Fig. 2-11 presents the corresponding tankage heat gain from structural radiation versus bulk mean tankage temperature. The net heat input from the heat exchangers was combined with the radiation heat gain, and an iterative solution for tankage temperature was obtained by 30-min increments. Initial tankage temperature was assumed to be $200^{\circ}R$, based upon the use toward the end of engine burn of hot-gas expulsion of propellants.

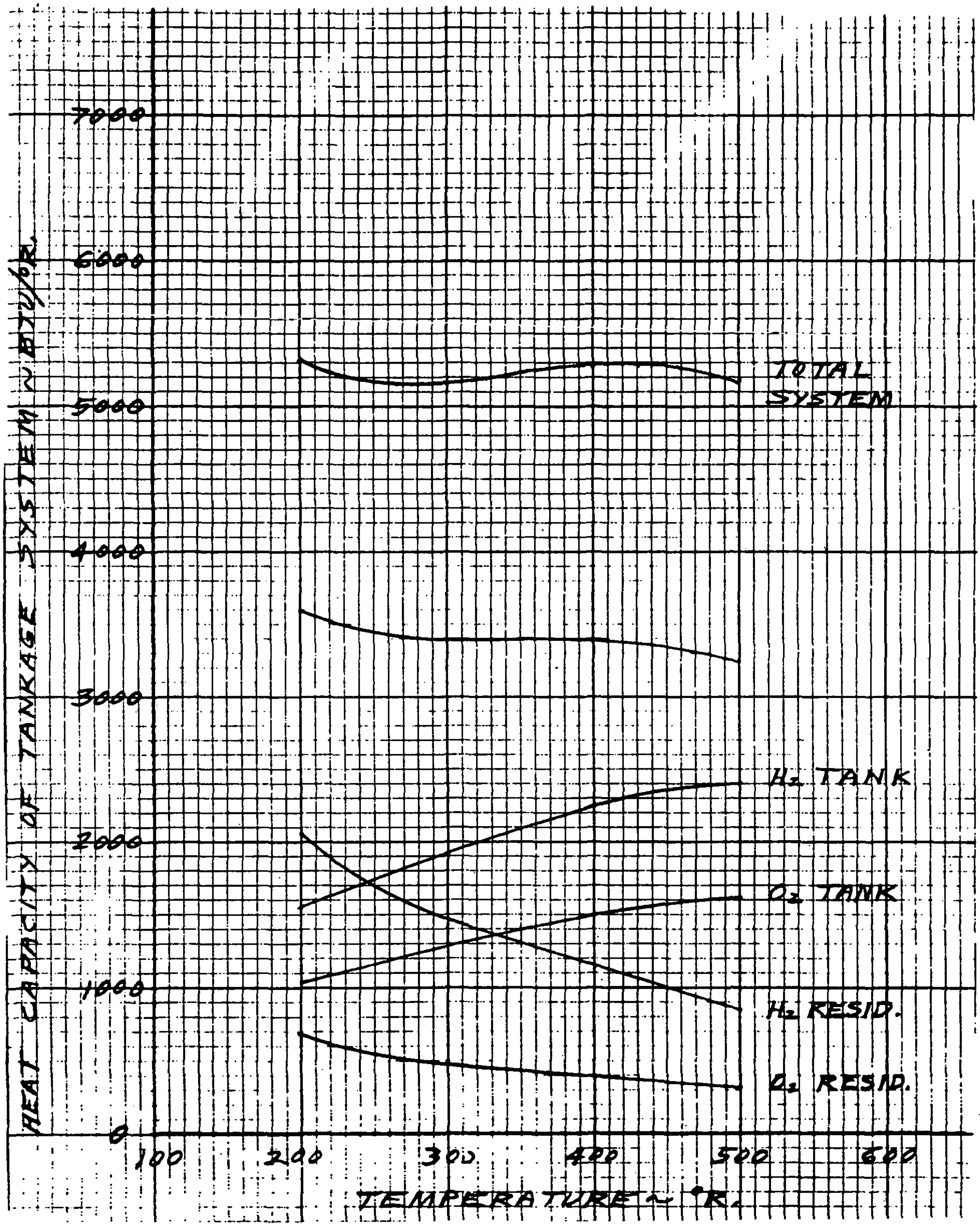


Fig. 2-10 Ascent Propellant Tankage System Heat Capabilities

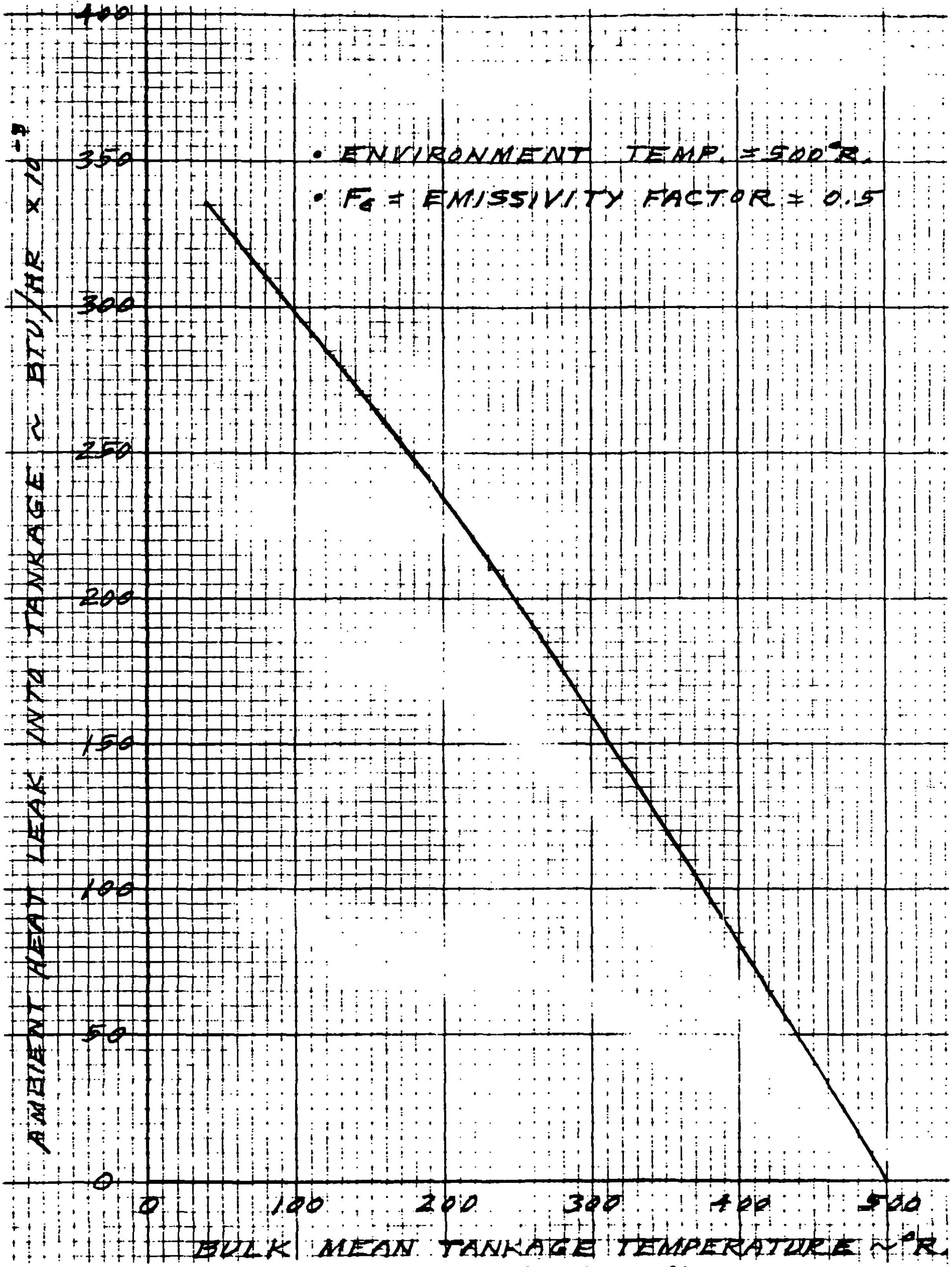


Fig. 2-11 Tankage System Heat Gain from Surroundings versus Tank Temperature for F_e = 0.5

Figure 2-12 shows resultant tankage temperature versus time from engine-burn shutdown for (1) the total tankage and residuals and (2) only the H_2 tankage and residuals. Maximum tankage temperature is seen to occur after 8 hours, if the complete tankage capacity is used; and after approximately 6.8 hours, if only the H_2 tankage capacity is used.

2.3.3.7 ACPS Conditioning Heat Exchanger and Tankage. The heat-transfer loop requirements to accommodate the very high heat-extraction rates for the ACPS propellant conditioning were evaluated. As noted previously, a maximum short-term heat-extraction rate of 4,795 Btu/sec is required for ACPS propellant conditioning: 3,520 Btu/sec for the H_2 propellant and 1,275 Btu/sec for the O_2. Since almost three-quarters of the total heat rate is required to be transferred by the H_2 loop, a relatively detailed analysis of that loop was made. It was assumed that the H_2 tankage temperature would remain essentially constant for short ACPS burns, and a tankage temperature of $500^{\circ}R$, was assumed. The H_2 temperatures to and from the H_2 conditioning heat exchanger were assumed to be $450^{\circ}R$ and $400^{\circ}R$, respectively:

a. Tankage Characteristics. Several assumptions were necessary to the determination of heat-transfer characteristics inside the H_2 tank:

(1) H_2 tankage heat capacity, including that of the residual GH_2, is concentrated in the tank walls.

(2) A distribution manifold delivers H_2 loop-returning fluid to five ducts that spray the H_2 onto the tank walls at appropriate velocity.

(3) Minimum thickness of the fluid flowing over the tank wall is one inch, and there is a 75 percent coverage of the entire internal wall.

Using these assumptions, it was determined that a mean fluid velocity of 150 ft/sec and flowrate of 18.75 lb/sec over the H_2 tank internal wall is required to transfer 3,520 Btu/sec, with an

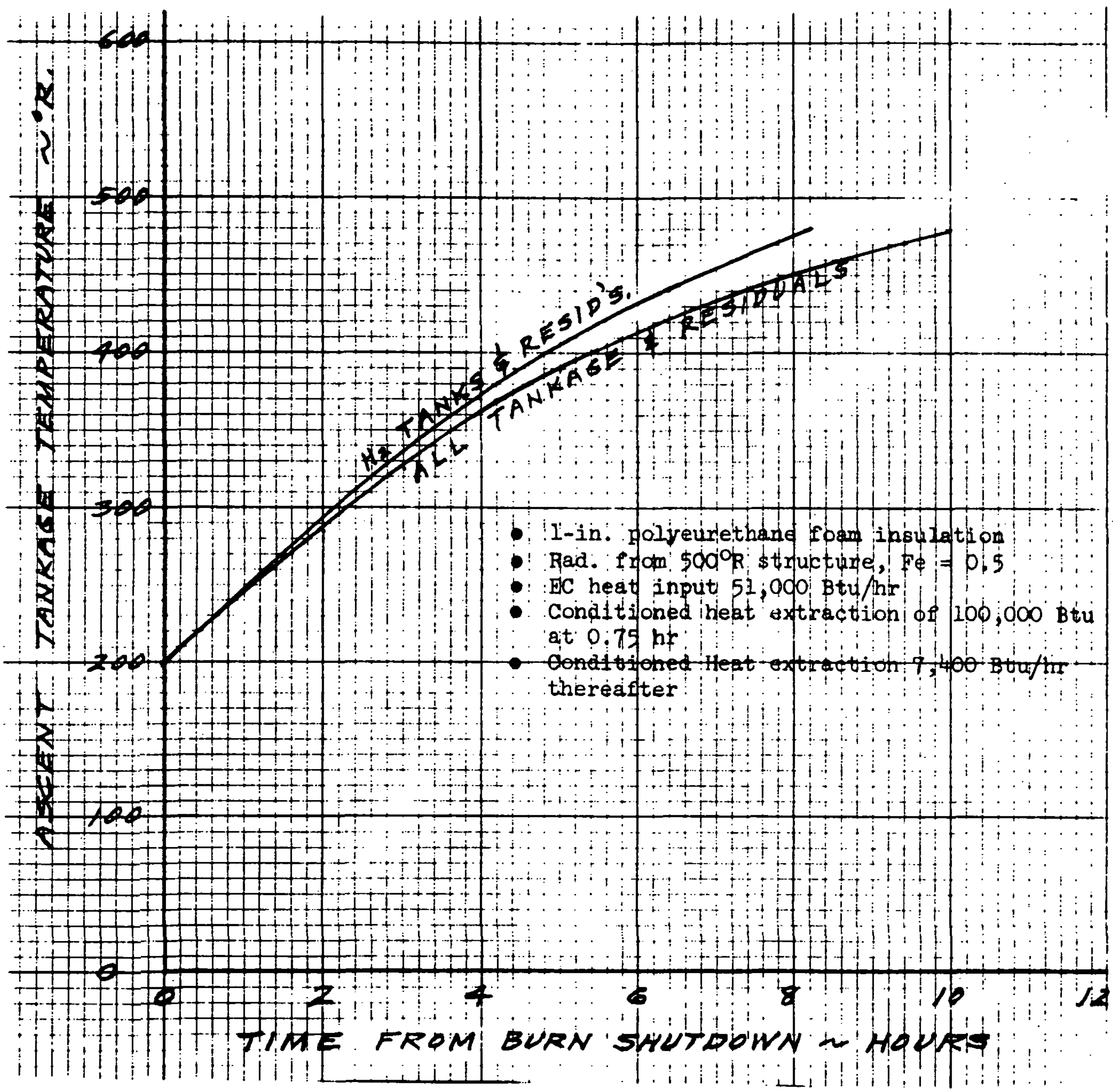


Fig. 2-12 Tankage Temperature vs Time

H_2 temperature change of $50^{\circ}R$ and an H_2-to-wall mean-temperature difference of $71^{\circ}R$. For this flowrate, the size and weight of H_2-loop gas compressor required to overcome the tank manifold and duct pressure drops was estimated as a function of the size of the manifold and ducts, up to 18-in. manifold diameter and 8-in. duct diameter. With these sizes, the tank internal ducting weight was estimated at 290 lb in aluminum, and the gas compressor was estimated to weigh 360 lb and require 1,075 hp at 50 percent efficiency.

b. <u>External Heat Exchanger Characteristics</u>. From the thermal requirements, a log mean-temperature difference of $200^{\circ}R$ was calculated for the external heat exchanger, leading to a required "UA" value of 3,400 Btu/hr $^{\circ}R$. Required heat exchanger effectiveness was 0.75. High-pressure ACPS H_2 propellant was assumed to flow inside the tubes of a shell-and-tube heat exchanger, and the H_2-loop fluid was assumed to flow around the tubes. Tube diameter was assumed to be 0.50-in.

An investigation of fluid velocity effects upon heat exchanger size showed little reduction in heat exchanger weight to be available, inasmuch as the H_2 velocities on the two sides were increased above 400 ft/sec (Mach = 0.1). Accordingly, fluid velocities of 400 ft/sec were used for the heat exchanger calculation, and the ratio of flow areas inside and outside the tubes was made appropriate to the ratio of fluid rates.

Using these conditions and assuming a heat exchanger length of 5 ft, an exchanger diameter of approximately 22 in. results, having a total estimated weight of 324 lb. The calculated pressure drop on the H_2-loop side would be in the area of 0.1 psi.

c. Summary of H_2 Loop System Characteristics. A summary of the estimated H_2-loop characteristics required for ACPS conditioning of H_2 propellant follows:

$$\dot{w}\,H_2 = 18.75 \text{ lb/sec}$$

$$V_{TNK_{H_2}} = 150 \text{ ft/sec}$$

$$V_{HX_{H_2}} = 400 \text{ ft/sec}$$

$$D_{TNK\ MAN} = 18 \text{ in.}$$

$$D_{TNK\ DUCTS} = 8 \text{ in.}$$

$$L_{HX} = 5 \text{ ft}$$

$$D_{HX} = 22 \text{ in.}$$

$$W_{TNK\ EQPT} = 290 \text{ lb}$$

$$W_{HX} = 324 \text{ lb}$$

$$W_{GAS\ COMPR} = 360 \text{ lb}$$

$$HP_{COMPR} = 1{,}075 \text{ hp}$$

These values are exclusive of external ducting between (1) the tank and heat exchangers and (2) of mounting structure. A quick estimate

indicates that the O_2-loop system weight would be approximately equal to that of the H_2-loop, and the O_2 compressor power would be about 50 percent lower.

2.3.3.8 EC/Electronics Heat Exchanger and Tankage. The heat-transfer loop requirements to accommodate the EC/electronics heat load were estimated. For this heat transfer task, the requirements were much less:

a. Heat load is a constant value of approximately 15 Btu/sec, as compared with the 3,520 Btu/sec required for ACPS H_2 conditioning reported in the foregoing paragraphs.
b. In this system, the H_2 loop was considered to perform the entire heat transfer task, since the H_2 tankage constitutes approximately 75 percent of the total tankage heat capacity.

For this analysis, it was assumed that the H_2 tank and residuals are maintained at $450^{\circ}R$ to accommodate a worst-case calculation. The H_2-loop fluid into and out of the external heat exchanger was taken as $450^{\circ}R$ and $475^{\circ}R$, respectively. EC/electronics cooling-fluid, Freon-21, was assumed to enter the heat exchanger at $610^{\circ}R$ and to exit at $500^{\circ}R$.

An analysis, similar to that for the ACPS conditioning system (as previously reported), was conducted. The rate of H_2 flow in the H_2-loop required to adequately transmit heat to the tank walls was found to be only on the order of 0.01 lb/sec. A calculation of heat exchanger tube-diameter effects indicated 0.125-in. tubes to be reasonable for the H_2 flow, and the flow area outside the tubes was adjusted to accommodate the Freon-21 flowrate. Corresponding velocities were approximately 160 ft/sec on the H_2 side and 10 ft/sec on the Freon-21 side. Based on these quantities, a heat exchanger of approximately 1-in. diameter and 10-in. length was estimated.

For this system, a simple longitudinal spray manifold in the H_2 tankage, about 0.5-in. diameter, would be adequate, and compressor requirements would be on the order of 1 hp.

Section 3

STUDIES APPLICABLE TO CURRENT SHUTTLE CONFIGURATION

3.1 INTRODUCTION

When the Cryogenic Cooling in Environmental Control Systems, Task IA of the Shuttle Cryogenic Supply Systems Optimization Study was approximately half way completed, the basic configuration of the Space Shuttle Vehicle was changed. The changes were a result of studies being performed by NASA and the alternate concepts contractors. Eventually a Space Shuttle configuration evolved that employed a resuable orbiter, orbiter main engine external tanks, and expendable solid rocket motors.

3.2 CURRENT SPACE SHUTTLE CONFIGURATIONS

A typical Space Shuttle configuration, as currently defined, is shown in Fig. 3-1. Unlike the earlier Shuttle configurations, the boost stages are not fully reusable. The solid rocket motors are separated at their depletion at about 132 seconds and a velocity of about 5513 ft/sec. The orbiter main engines, which are ignited on the ground, continue to withdraw the oxygen and hydrogen from the external ascent tank. After injection into an elliptical orbit the main engines are shutdown and the ascent tank is jettisoned and de-orbited prior to the time the orbiter is injected into its circular orbit by the orbit maneuver propulsion system.

The major subsystems that contributed to the Cryogenic Cooling in Environmental Control Systems for the Phase B Shuttle configurations are generally not applicable to the current Shuttle configuration. The Orbit Maneuver Propulsion System, (OMPS), the Reaction Control System (RCS), and the Auxiliary Power System (APU) no longer employ cryogenic fluids but now employ fluids that are liquids at normal room temperatures. These systems and the fluids they use are shown in Fig. 3-2. The only system that still employs cryogens is the electrical power supply fuel cell modules.

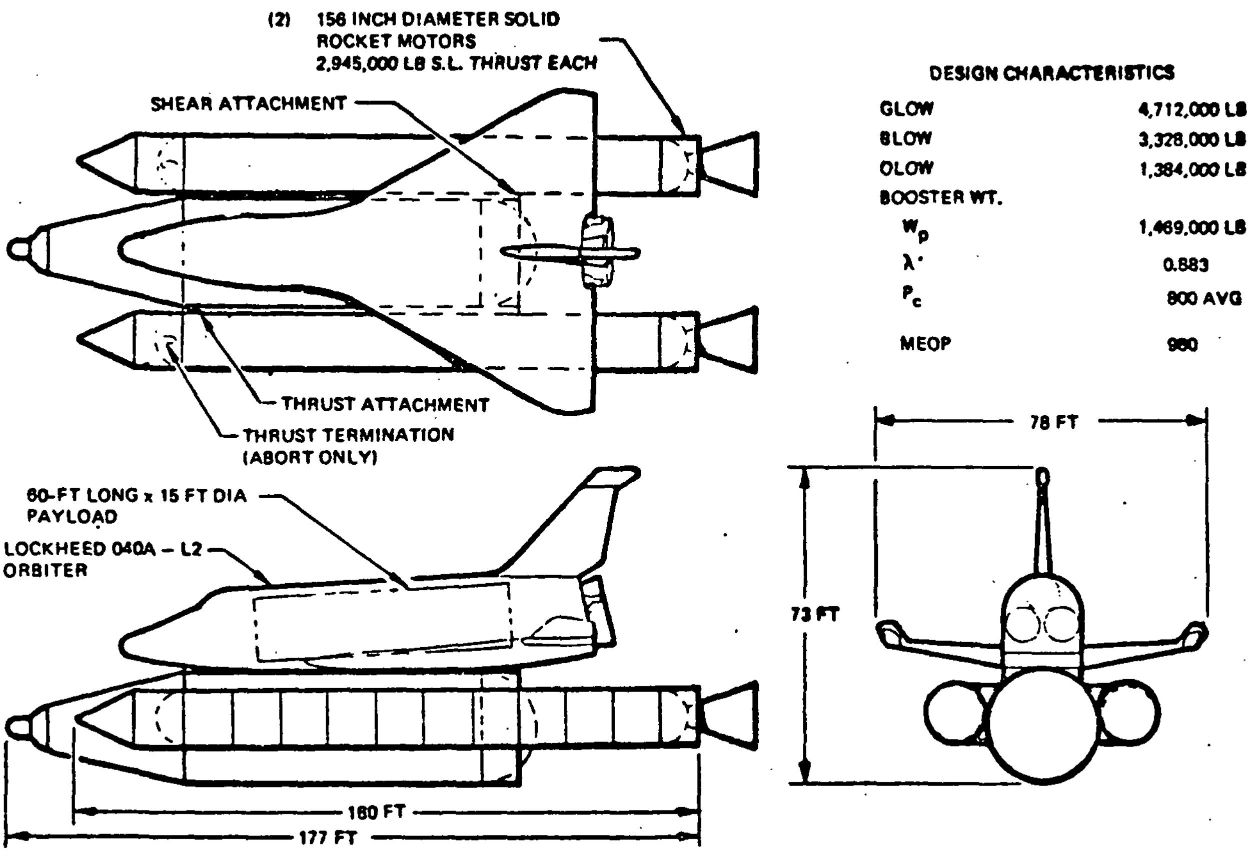


Fig. 3-1 Typical Current Space Shuttle Configuration

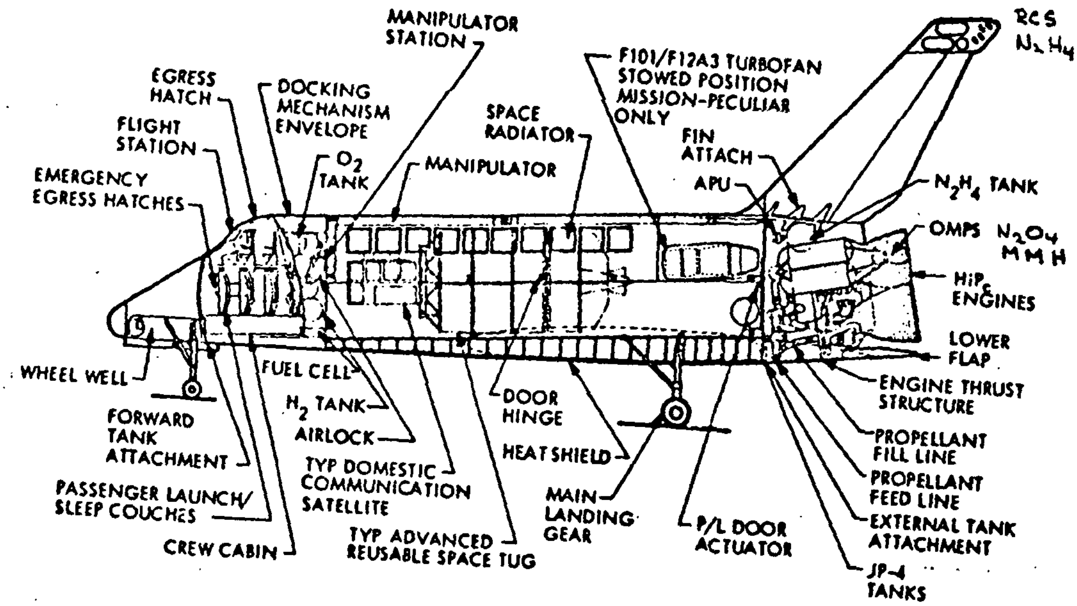


Fig. 3-2 Typical Orbiter Inboard Profile

The environmental control system and the radiators are essentially the same as before and the comments in Section 2.2 are applicable. The heat loads are essentially unchanged; however the availability of cryogenic heat sinks is drastically reduced. As a result, studies regarding environmental control systems were directed toward the broader aspect of systems environmental thermal control. Many studies are applicable to this current configuration and some of the studies initiated earlier for the Phase B Shuttle configuration are described in this section.

3.3 STUDIES

3.3.1 Freon 21/Cryogenic Heat Exchanger

Early in the Task 1A effort a study for Freon 21-hydrogen and Freon 21-oxygen heat exchangers was initiated with AiResearch Manufacturing Company. A range of parameters (see Table 3-1) for which the heat exchangers were to be evaluated was established based on the Phase B Shuttle configurations and subsystem operating characteristics. The cryogenic fluid conditions were based on using the hydrogen and oxygen from either subcritical or supercritical storage vessels and on using appropriate pumps. These conditions were primarily tailored to an integrated OMPS - ACPS system that employed pumps, heat exchangers, and accumulators. However, the range of parameters was broad enough and the analyses basic enough to permit the study to be applicable to fuel cell reactant supply conditioning and to Cryhocycle systems. Therefore, the results of the study are included in this section.

3.3.1.1 Core Construction. All units utilize stainless steel, shell-and-tube matrixes of brazed and welded construction. Furthermore, in all cases, the Freon 21 is multipassed outside of the tubes in overall counterflow arrangement. Figure 3-3 illustrates the construction of a typical unit. Because of pressure containment considerations for fluids above 450 psia, (and zero leakage requirement), tubular construction was selected over plate-fin. Another contributing influece was the typical small size of the units. Stainless steel was selected over aluminum for increased reliability and greater

Condition Number	1A	1A*	1B	1B*	2A	2B	3	4	5	6	7	7*
Problem Statement												
Hot Side												
Fluid	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21
Flow, lb/hr	17.1	244.0	17.1	244.0	488.0	488.0	1220.0	488.0	1220.0	1710.0	1220.0	244.0
T_{IN}, °R	660.0	660.0	660.0	660.0	660.0	660.0	660.0	660.0	660.0	660.0	660.0	660.0
T_{OUT}, °R	500.0	500.0	500.0	500.0	500.0	500.0	500.0	500.0	500.0	500.0	500.0	500.0
P_{IN}, psia	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0
ΔP, psi (matrix only)	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0
Cold Side												
Fluid	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2
Flow, lb/hr	0.40	5.7	0.612	8.75	11.4	17.5	60.7	24.9	62.2	87.1	64.1	12.82
T_{IN}, °R	40.0	39.0	200.0	200.0	40.0	200.0	39.0	39.0	39.0	39.0	39.0	39.0
T_{OUT}, °R	500.0	500.0	500.0	500.0	500.0	500.0	250.0	250.0	250.0	250.0	250.0	250.0
P_{IN}, psia	450.0	450.0	450.0	450.0	450.0	450.0	25.0	450.0	450.0	450.0	1000.0	1000.0
ΔP, psi (matrix only)		0.1		0.1	0.5	5.0	2.0	0.5	0.5	0.5	0.1	0.1
General Information												
Heat rejection, Btu/hr	700.0	10000.0	700.0	10000.0	20000.0	20000.0	50000.0	20000.0	50000.0	70000.0	50000.0	10000.0
Maximum TCR (hA_I/hA_O)	0.95	0.95	3.0	3.0	0.95	3.0	0.95	0.95	0.95	0.95	0.95	0.95
Design												
Effective Matrix Parameters												
Tube length, in.		6.5		6.8	8.9	12.1	4.7	6.3	6.4	5.0	3.8	3.7
Outside-the-tube length, in.		0.410		0.276	0.374	0.270	0.878	0.317	0.553	0.741	0.78	0.298
No-flow length, in.		0.313		0.299	0.361	0.318	0.959	0.398	0.611	0.94	0.894	0.368
Tube diameter, in.		0.1		0.1	0.1	0.1	0.1	0.100	0.100	0.100	0.1	0.1
TCR (hA_I/hA_O)		0.736		0.476	0.697	1.612	0.4	0.713	0.693	0.720	0.647	0.626
Tube weight, lb		0.12		0.02	0.05	0.04	0.12	0.03	0.07	0.11	0.08	0.02
Overall Core Parameters												
Shell length, in.		6.8		7.1	9.2	12.4	5.0	6.6	6.7	5.3	4.1	4.0
Shell diameter, in.		0.5		0.45	0.5	0.45	1.25	0.5	0.75	1.25	1.1	0.5
Unit weight, lb	<0.1	0.2	<0.1	0.2	0.3	0.3	0.6	0.2	0.4	0.6	0.5	0.4

TABLE 3-1 - RANGE OF PARAMETERS FOR HEAT EXCHANGERS

Condition Number	8	8*	9	10	11	12	12*	13	14	15	16	17A	17B
Problem Statement													
<u>Hot Side</u>													
Fluid	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21	F-21
Flow, lb/hr	1220.0	244.0	488.0	1220.0	1220.0	1220.0	244	567.0	567.0	567	1220.0	854.0	854.0
T_{IN}, °R	660.0	660.0	660.0	660.0	660.0	660.0	660.0	660.0	660.0	400.0	660.0	660.0	660.0
T_{OUT}, °R	500.0	500.0	500.0	500.0	500.0	500.0	500.0	300.0	300.0	250.0	500.0	500.0	500.0
P_{IN}, psia	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	450.0	300.0	300.0	300.0
ΔP, psi (matrix only)	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	0.5	3.0	3.0	3.0
<u>Cold Side</u>													
Fluid	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2	O_2	O_2	O_2
Flow, lb/hr	67.2	13.44	11.3	28.3	28.4	28.6	5.72	28.3	28.6	24.9	322.0	234.0	745.0
T_{IN}, °R	39.0	39.0	39.0	39.0	39.0	39.0	39.0	39.0	39.0	39.0	165.0	180.0	350.0
T_{OUT}, °R	250.0	250.0	500.0	500.0	500.0	500.0	500.0	500.0	500.0	250.0	500.0	500.0	500.0
P_{IN}, psia	2000.0	2000.0	25.0	25.0	450.0	1000.0	1000.0	25.0	1000.0	450.0	1000.0	1000.0	1000.0
ΔP, psi (matrix only)	0.1	0.1	2.0	2.0	0.5	0.25	-	0.05	0.01	0.5	5.0	5.0	10.0
General Information													
Heat rejection, Btu/hr	50000.0	10000.0	20000.0	50000.0	50000.0	50000.0	10000.0	50000.0	50000.0	20000.0	50000.0	35000.0	35000.0
Maximum TCR (hA_I/hA_O)	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.11	0.11	No limit	2.0	2.3	No limit
Design													
<u>Effective Matrix Parameters</u>													
Tube length, in.	4.0	4.1	9.2	6.7	9.1	8.6	-	6.6	≅6.0	8.7	14.3	15.4	9.2
Outside-the-tube length, in.	0.635	0.327	0.432	0.753	0.601	0.547	-	2.0	≅1.35	0.476	0.423	0.407	0.579
No-flow length, in.	0.687	0.273	0.469	0.856	0.662	0.595	-	2.0	≅1.35	0.516	0.526	0.431	0.631
Tube diameter, in.	0.1	0.1	0.1	0.1	0.1	0.1	-	0.1	0.1	0.1	0.1	0.1	0.1
TCR (hA_I/hA_O)	0.673	0.756	0.385	0.376	0.736	0.720	-	0.1	0.11	0.649	0.829	0.737	0.899
Tube weight, lb	0.07	0.01	0.07	0.17	0.12	0.11	<0.02	1.1	0.75	0.07	0.12	0.10	0.13
Overall Core Parameters													
Shell length, in.	4.3	4.4	9.5	7.0	9.4	8.9	-	6.9	6.3	9.0	14.6	15.7	9.5
Shell diameter, in.	0.8	0.45	0.6	1.1	0.75	0.75	-	2.5	1.60	0.7	0.7	0.6	0.8
Unit weight, lb	0.4	0.1	0.4	0.7	0.6	0.6	<0.15	3.5	2.3	0.4	0.6	0.5	0.65

TABLE 3-1 (CONT'D) - RANGE OF PARAMETERS FOR HEAT EXCHANGERS

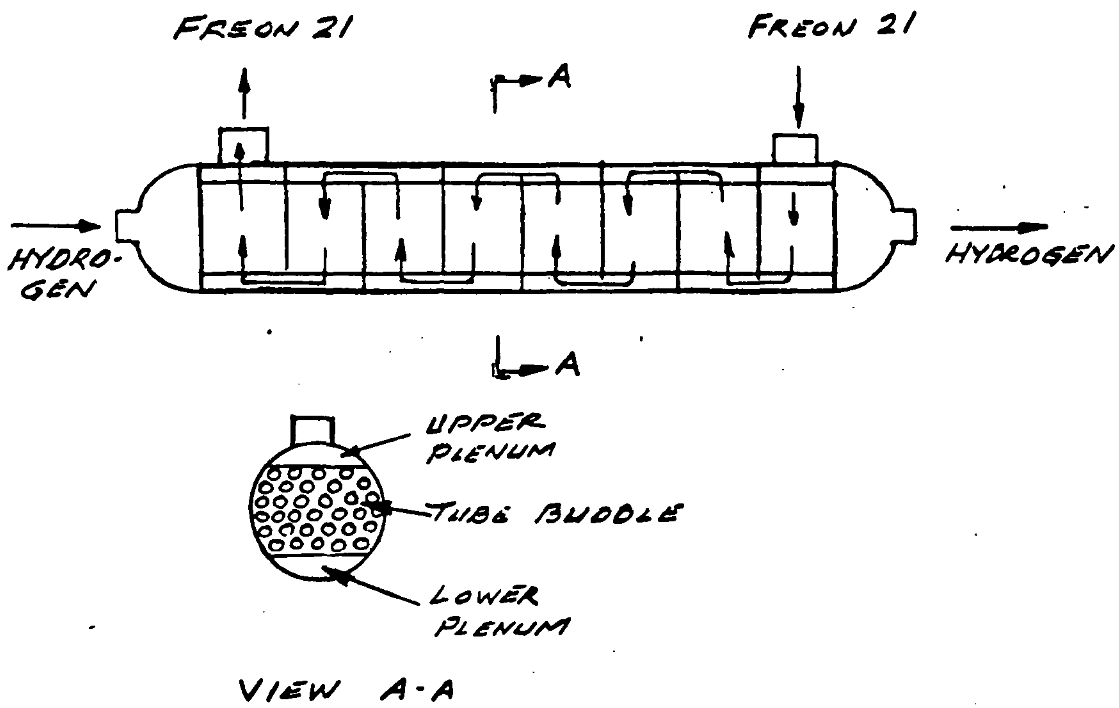


Fig. 3-3 Typical Heat Exchanger

ease of manufacturing small-diameter, closely-packed matrixes. Aluminum has better strength-to-weight characteristics than nickel or stainless steel and has a definite weight advantage for all structures above minimum gauge. For these units, however, all items such as tube wall thickness and tube spacing are all at the minimum gauge. Because of considerations such as braze penetration, the minimum stainless steel tube wall thickness is 0.006-in. compared to 0.016-in. for aluminum in a typical 0.100-in. outside diameter tube. This alone overshadows the 2.86 weight advantage of aluminum. In addition, in aluminum more tubes are required for the same pressure drop since the tube inside diameter is 0.088-in. and 0.068-in., respectively, and free flow area varies with the square of inside diameter. Counterflow designs were generally dictated by the heat transfer requirements.

3.3.1.2 Discussion. The most important limiting side condition for all Freon 21-to-cryogenic fluid heat exchangers was the maximum thermal conductance ratio (TCR) permissible to preclude freezing of Freon 21. The freezing problem is discussed in more detail in subsection 3.3.1.4. Assuming a Freon 21 freezing point of 249°R, a minimum wall temperature of 275°R was deemed permissible. Therefore, for counterflow units with a Freon 21 outlet temperature of 500°R and a cryogen inlet temperature of 39°R, one finds the maximum TCR as follows:

$$hA_I\,(T_w - T_c) = hA_o\,(T_h - T_w) \qquad \text{(cold fluid inside the tubes)}$$

$$TCR = \frac{hA_I}{hA_o} = \frac{T_h - T_w}{T_w - T_c}$$

$$TCR = \frac{500 - 275}{275 - 39}$$

$$TCR = 0.954$$

For the two conditions with a 300°R Freon 21 outlet temperature and a 39°R cryogen inlet temperature, the maximum permissible TCR is 0.106 (or, in other words, the cryogen must have a controlling thermal resistance). For this

reason, the design for Conditions 13 and 14 (see Table 3-1) are very large relative to all other units, and the usable cryogen pressure drop is small for all units.

Secondary design considerations were to create designs with desirable shape that are packageable and that have predictable performance in severe acceleration fields. Shape and packageability are very significant for these small units, because the wrap-up weight dominates through items such as the shell, header plates, and baffles. In many instances, reducing matrix weight by increasing the hydrogen pressure drop actually causes a total weight increase. For very low velocity, laminar flow, the cryogen heat transfer coefficient is sensitive to the local acceleration field. Therefore, to ensure acceptable operation during periods of sustained acceleration and deceleration, all designs were limited to turbulent flow inside the tubes. Furthermore, the tubes were ring-dimpled to decrease weight and to ensure predictable performance.

3.3.1.3 Off-Design Point Performance. The most critical factor for off-design point performance is the Freon 21 outlet temperature. Assuming only a reduction in Freon 21 flow, it may be readily shown that the Freon 21 outlet temperature will decrease and the TCR will increase. The combination of these two effects will dramatically impose a wall-freezing problem for all units with a 39° or 40°R cryogen inlet temperature. The governing equations for this phenomena are the effectiveness-Ntu equation for a counterflow heat exchanger, and the variation in Freon 21 heat transfer coefficient with flow (Berglin's published data for high Prandtl number flow outside of equilaterally-spaced tube banks is representative). If significant reductions in Freon 21 flows are anticipated, then the maximum permissible TCR at the thermal energy design point must be reduced. Assuming a fixed design point (i.e., UA requirement), and a fixed outside-the-tube heat transfer coefficient, the weight of a given design varies with the allowable TCR in the following manner:

$$\text{Weight} = \frac{1 + \text{TCR}}{\text{TCR}}$$

For TCR substantially less than one, the relationship becomes almost direct. For example, take the 500°F, Freon 21 outlet temperature designs with a maximum permissible TCR of 0.95. If, in order to permit operation at lower Freon 21 flow, the maximum permissible TCR is reduced by one-half to 0.475, the weight of the unit must be increased by a factor of 1.5. Alternative solutions would be to greatly increase the Freon 21 pressure drop, which would reduce the total weight, would still experience the above weight penalty relationship for low Freon 21 flow operation, or would establish an acceptable control system. The best method of control would be to bypass cryogen flow at low-sensed Freon 21 outlet temperature.

3.3.1.4 Potential Freezing Problems

Design Criteria Selection. Within any two-fluid heat transfer matrix, a potential freezing problem can exist if the cold fluid temperature is below the hot fluid freeze point in all or part of that matrix. For liquids that undergo large changes in visocosities at low temperature, such as MIL-L-7808 synthetic lubricating oil, the quasi freezing problem of progressive congealing may also occur, even at temperatures substantially above the freeze or pour point of the liquid. It is necessary and sufficient that the hot fluids wetted wall temperature be maintained above an established critical temperature. Because of the relatively mild variation of Freon 21 viscosity with temperature near its freeze point, 249°R, an arbitrary minimum permissible wall temperature of 275°R was selected as a design criteria to preclude progressive congealing or freezing.

Wall Temperature. Having established a design criteria, it is necessary to investigate the wall temperature predicted equation and its ramifications for a given design.

By applying the convection heat transfer equation to any boundary area between the cryogenic fluid and the Freon 21 in a typical compact matrix, one obtains a steady-state solution of the following equation:

$$hA_{COLD}\,(T_{WALL} - T_{COLD}) = hA_{HOT}\,(T_{HOT} - T_{WALL})$$

By defining the thermal conductance ratio, TCR, as a cold side heat transfer conductance divided by the hot side heat transfer conductance and rearranging the above expression, the common wall temperature prediction equation is obtained

$$T_{WALL} = \frac{T_{HOT} + TCR\ T_{COLD}}{1 + TCR}\ ,\quad TCR \triangleq \frac{hA_{COLD}}{hA_{HOT}}$$

It is readily apparent that the local wall temperature depends upon the special fluid temperature distribution within the matrix and the local fluid heat transfer conductances. By developing a suitable nodal model, the most accurate method of predicting wall temperatures is through computerized finite difference techniques, which include the second order temperature effects on local heat transfer conductances. AiResearch has standard working computer programs that have been written for this type of analysis; however, such methods are too time-consuming, in both man hours and in computer time, to be rigorously employed to each candidate solution in a parametric study. An alternate first-order approximation can be used with acceptable accuracy for design studies.

The basic assumption in this approximation is that the thermal conductance ratio is constant and uniform throughout the matrix. This assumption is quite good, since the heat transfer coefficient is only a mild function of temperature. Predicated on this assumption of virtually constant thermal conductances, the spacial temperature distribution is theoretically predictable for a given flow arrangement. It is now possible to investigate the general temperature distribution and to calculate the maximum permissible thermal conductance ratio for a design point or design conditions. This calculated value then becomes a selection criterion for all candidate solutions. For example: with a pure counterflow arrangement, if one inserts the critical temperature for the wall temperature in the previously-derived equation and calculates the TCR required to prevent freezing at various points along the flow path, it will be found that the maximum permissible thermal conductance ratio will occur at the hot fluid outlet and, synonymously, at the cold fluid inlet. If one designs the heat exchanger to prevent this point from freezing, the

heat exchanger will not freeze at any point within the matrix. The prediction of temperature distributions in various flow arrangements can be found in any basic text book on heat transfer or heat exchanger designs.

Thermal Conductance Ratio Impact on Heat Exchanger Sizing. Having established the significance of a thermal conductance ratio on wall temperature, it is necessary to determine its impact on heat exchanger size. For a given design condition and flow arrangement, there is a specific total thermal conductance, UA, that is required. By definition

$$\frac{1}{UA} = \frac{1}{hA_{COLD}} + \overset{0\approx}{\cancel{\frac{x}{KA}}} + \frac{1}{hA_{HOT}}$$

which becomes, after rearrangement and substitution,

$$UA = \frac{hA_{COLD}}{1 + TCR} = \frac{TCR\ hA_{HOT}}{1 + TCR}$$

Note that the area in this equation is directly proportional to heat exchanger size, weight, and volume. Optimization procedures are instigated with the general purpose of maximizing U within the confines of allowable pressure loss in order to minimize the area, A. Consider any arbitrary design point that has a predetermined maximum permissible TCR. The obvious impact on any design by the first term in the above expression is a maximum limit on hA_{COLD}, which in turn affects hA_{HOT} through the TCR definition. This imposes a restriction on total heat exchanger size and weight. The limit on the hot magnitude of the hot and cold side conductances has been uniquely determined by the wall temperature side condition. This restriction is independent of the pressure drop allocation, so that the allowable pressure drop on either side may not be fully utilized.

Off-Design Point Performance. Having established a theoretical background for evaluating heat exchangers with the freezing problem, off-design point performance can now be investigated. Changes in fluid flow of either or both sides will necessarily change both the estimated thermal conductances and the

temperature distribution within the matrix. These effects may be evaluated for a specific operating point using the methodology and equations derived in the previous paragraphs. Depending upon the design point condition, small decreases in hot fluid flow may result in large changes in outlet temperature, with the ultimate effect on a freezing condition in at least part of the heat exchanger. In fact, for any design unit, a sufficiently large decrease in hot fluid flow at fixed cold fluid flow will result in a predicted outlet temperature below the freezing point of the hot fluid. Therefore, the range of hot fluid variation and/or cold fluid variation must be bounded by system requirements and the unit designed accordingly. For example, consider a pure counterflow design with a high flow operating condition, which imposes a maximum UA requirement. If this unit operated at, say, half flow, the resultant hot fluid outlet temperature will be lower because of the influence of capacity rate ratio on effectiveness, all other things being constant. This reduced hot fluid outlet temperature could, and probably would, impose a much more severe limitation of the maximum permissible thermal conductance ratio. Therefore, to ensure performance at this condition, it would be necessary to reduce the cold side thermal conductance appreciably and, therefore, necessitate a much larger unit at the design condition than would be required by its own TCR requirement. A rough first approximation to investigate such possibilities in a submitted design would be to assume that the thermal conductances were independent of flow and to estimate the outlet temperature of the hot fluid at an off-design condition. From this estimate one can evaluate the new thermal conductance ratio required to preclude freezing and compare it with the designed value. If it is below the designed value, then a new design must be generated with a substantially decreased cold side thermal conductance to protect the heat exchanger over desired hot flow range. In all of the submitted designs this would impose a weight penalty that can be evaluated from the UA relationship. A more detailed approach would require a calculation of the new thermal conductances at reduced flow, to illustrate the impact of the new temperature distribution as well as the new mass velocity through the unit on the predicted thermal conductances.

Physical Design Considerations. Having provided a background on thermal conductance ratio and its relationship to the freezing and a general theoretical approach to the freezing problem, it is now possible to discuss the practical aspects of heat exchanger design with regard to the freezing problem. The following is a discussion of the practical design considerations employed in the parametric study.

Flow Distribution. For all units, the frictional drop through the matrix is maintained at least a factor of 10 larger than the incident velocity head and matrix flow acceleration term. This provides assurance that the fluids remain evenly distributed throughout the matrix. Flow maldistribution upsets the temperature distribution within matrix and may cause the following problems:

(a) Reduced overall heat transfer performance

(b) Increased unit pressure drop

(c) Excessively hot or cold zones within the matrix which may cause

- (1) High thermal stresses
- (2) Structural weakening of the matrix
- (3) Undesirable local phase changes in the fluid streams

Finned Surfaces. Finned surfaces are used to balance the thermal conductances to achieve a compact-lightweight matrix, and permit the best utilization of the available pressure head. Since the fins may support an appreciable temperature gradient and often suffer a quasi-stagnation area near the fin root, both the root and tip temperatures must be investigated to avoid the adverse effects of temperature extremes. For example, when freezing or congealing is in question, it is often necessary to avoid fins on the hot fluid side. Although the conductance ratio would tend to establish a lower "mean" wall temperature, the low root temperature and its quasi-stagnation area permit the initiation of freezing that may progressively fill the fins. This causes a large loss in effective heat transfer area and freeze flow area which, in turn, means lower heat transfer performance and high pressure drop.

Flow Regime. The operating flow regime of heat exchangers can exert a large influence on the predictability of heat exchanger designs. At large mass velocities and high Reynold's numbers (in excess of $5\ (10)^5$), as commonly found with high pressure hydrogen gas units, it is often difficult to achieve a sufficiently high fractional loss to achieve flow stability. However, the heat transfer coefficients are typically so high that the gravitational field has a negligible effect on predicted performance. For liquids undergoing large changes in viscosity with temperature, it is beneficial to try to achieve turbulent flow over the probable operating flow range. The frictional pressure loss is directly proportional to the viscosity in laminar flow, whereas in turbulent flow frictional pressure drop may vary by only viscosity to 0.2 power. Obviously, the impact of viscosity is greatly reduced for turbulent flow. At very low mass velocity and laminar-type flow, the heat transfer coefficient can be so small that it becomes sensitive to imposed gravitational fields. A final consideration is the predictability of performance over a specified flow range. If a transition zone is transversed in off-design-point operation, special design problems such as freezing may become evident. To conclude, predictable and stable heat exchanger performance can be augmented by judicious selection of the design flow regime. Off-design operating range can also be increased by appropriate flow regime selection.

Geometry Selection. Although there are numerous varieties and types of heat transfer matrixes, the most common for general compact-type application are the plate-fin and the small diameter tube bundle. The plate-fin type matrix offers the following:

(a) High heat transfer area density per unit following

(b) High performance surfaces via boundary-layer interrupting, off-set fins

(c) Ease in balancing thermal conductance and overall geometry by varying the surface combinations

Small diameter tubular matrixes are not as good as the plate-fin type in the above categories, but they do offer other advantages:

(a) Excellent pressure containment characteristics particularly in the classic shell-and-tube version

(b) Usually a reduced total length of brazed-joint, fluid interface, and therefore a reduced liklihood of inter-fluid leakage

(c) By proper design, good control of freezing or congealing may be achieved

The advantages of shell-and-tube matrixes for the Freon 21-to-cryogenic fluid application outweighed those of the plate-fin type. By flowing hot fluid outside the tubes, total tube blockage was eliminated. If a tube were fully blocked, high tube-to-tube thermal stresses could arise.

3.3.2 Mission Heat Profile Studies

A detail mission heat load profile was generated and is summarized in Table 3-2 for the three phases of flight: prelaunch/ascent, orbital, and re-entry/landing. This includes heat from electrical power, cabin wall and window heating, metabolic and other chemical heat sources, fuel cells, APU and hydraulic pumps, and ambient heating. The values currently obtained are approximate and in some cases are crude estimates.

The major loads are reasonably well known and it is not expected that major changes will result as more data become available. The heat loads to the cabin and the heat picked up by the hydraulic system are not known at this time, so estimates for these values have been included. These heat loads are used to size the various cooling systems and to evaluate the need for cooling during various phases of flight.

From preliminary observation it appears that adequate cooling can be obtained during prelaunch/ascent and orbital phases without utilizing dedicated expendable coolants (hydrogen, in particular). During ascent, the cooling may be accomplished by several means. The system may be simply allowed to heat up until the radiators can be deployed, or generated water can be sublimated after achieving high altitude, or heat may be transferred to the ascent tank

Table 3-2

ESTIMATED HEAT LOADS

Phase	Time (Hr:Min:Sec)	ΔT (Hr)	Electrical (Btu/hr)	Cabin Wall (Btu/hr)	Metabolic (Btu/hr)	Fuel Cell (Btu/hr)	APU & Pump (Btu/hr)	Ambient (Btu/hr)	Total Rate (Btu/hr)	Total Load (Btu)
Ground Hold	-00:10:00 00:00:00	0.167	24,200	5,260 (1)	2,000	14,900	153,000	0	199,360	33,121
Ascent	00:00:00 00:02:39 00:09:15	0.044 0.110	24,100 29,200	13,000 13,000	2,000 2,000	14,900 20,600	153,000 153,000	0 0	207,000 217,800	9,100 23,960
Orbit	00:09:15 01:51:00 167:50:00	1.7 159.0	24,000 18,300	0 0	2,000 2,000	16,900 12,870	0 0	0 0	42,900 33,170	72,900 5,502,400
Reentry & Landing	167:50:00 168:38:00 169:11:00 169:23:05	0.8 0.55 0.20	22,550 22,800 26,350	0 98,000 (1) 98,000 (1)	2,000 2,000 2,000	13,900 14,200 16,200	153,000 153,000 153,000	0 190,000 (1) 189,000 (1)	191,000 480,000 485,000	153,300 262,950 97,162

(1) Rough estimates

cryogens and residuals prior to the time they are dropped, or any combination of these methods may be employed.

During the orbital phases it appears that radiators, supplemented by water sublimation and fuel cell cryogen heating, will provide sufficient cooling. It does not currently appear that fuel cell cryogen heating is a must to obtain a proper heat balance and temperature balance; however, this conclusion depends upon type and design installation of the radiators. Furthermore, the use of cryogen heating can be of aid in preserving water for later high heating periods and may provide supplemental cooling at times when venting of vapors is not desirable.

During reentry and aerodynamic flight, several methods of cooling are available. Water can be sublimated or boiled during most of the reentry periods, dedicated hydrogen can be used and ram air cooling can be used.

To help determine what heat load should be imposed on the cooling system, estimates have been made of how much heat can be rejected to the discardable droptanks during ascent and to ambient air during the return portion of the flight. These results are summarized in the following sections.

3.3.2.1 Ascent Cooling. A preliminary analysis was conducted to determine if the ascent propellants could be used as a heat sink. The analysis considered four early mission phases and associated heat loads, as follows:

Phase	Time (Min.)	Total Heat Load (Btu)
Ground Hold	10	33,121
Boost & Coast	2.67	9,100
Orbit Injection	6.58	23,960
Pre-tank Drop	32.75	23,450

For the ground hold, boost and coast, and orbit injection phases, the analysis considered the tanked propellant heat capacities only, and allowed an LO_2 temperature rise three times the LH_2 temperature rise for the purpose of equalizing the respective vapor pressure increases. Under these conditions,

the temperature rise due to the imposed heat load, from ground hold through orbit injection, was estimated to be

$$\Delta T_{H_2} = 0.10°R$$

$$\Delta T_{O_2} = 0.31°R$$

If all the heat were imparted to the tanked LH_2 exclusively, the corresponding temperature rise would be approximately

$$\Delta T_{H_2} = 0.5°R$$

For the time interval between orbit injection burnout and tank drop, several alternate sinks were considered:

- The heat of vaporization of the trapped LO_2 in the orbiter and the heat capacity of the vapor
- The heat of vaporization of the trapped LH_2 in the orbiter and the heat capacity of the vapor
- The heat capacity of the residual GH_2 in the H_2 tank. (No heat capacity was considered to exist in the GO_2 because of its high temperature.

The estimated capacities of the propellants assumed to be trapped in the orbiter far exceeded the 23,450 Btu heat load. These calculated capacities were

Trapped LO_2:	170,000 Btu
Trapped LH_2:	101,000 Btu

The H_2 tank residual gas has the capacity to absorb the 23,450 Btu heat load at a GH_2 use rate of 0.8 pound per minute. Since the required average tank evacuation rate during the 32.75-minute pre-tank drop period is 32.3 pounds

per minute, the tank residual gas far exceeds the heat sink requirements during this period.

3.3.2.2 Ram Air Cooling. To better define how much dedicated fluid would be required during reentry, an investigation was made to determine the capability of achieving rejection of the EC/LSS heat to ram air during descent by means of passing ram air between the folded and stowed space radiators. The possibility of cooling the hydraulic oil only, by means of a fin-and-tube, oil-to-air heat exchanger, was also explored.

Analysis was based on the following ram air conditions:

Time from 4000,000 Ft. (Min.)	Altitude (Ft.)	Mach Number	Std. Day Ram Air Stagnation Temperature (°F)
31.03	125,000	3.3	990
32.497	102,000	2.8	640
34.16	78,000	1.4	85
35.83	56,000	1.0	8
37.01	40,000	0.8	-20
40.40	20,000	0.5	10
45.00	0	0.3	69

It was assumed that the cooling air temperature available at the heat exchange surface would be equal to the total (stagnation) temperature, and the altitudes above 56,000 feet were eliminated from consideration.

For purposes of evaluating ram air cooling of the coolant loop Freon in the folded and stowed space radiators, it was assumed that the Freon inlet temperature would be 80°F and the air temperature rise would be 10°F, with a desired Freon outlet temperature of 30°F. Under standard day conditions this is impossible to attain at sea level.

Figure 3-4 shows the air flow area required for Mach 0.3 flow in the air flow path between the folded radiator panels, based on the altitude air density, the applicable heat exchange rate, and an air temperature rise

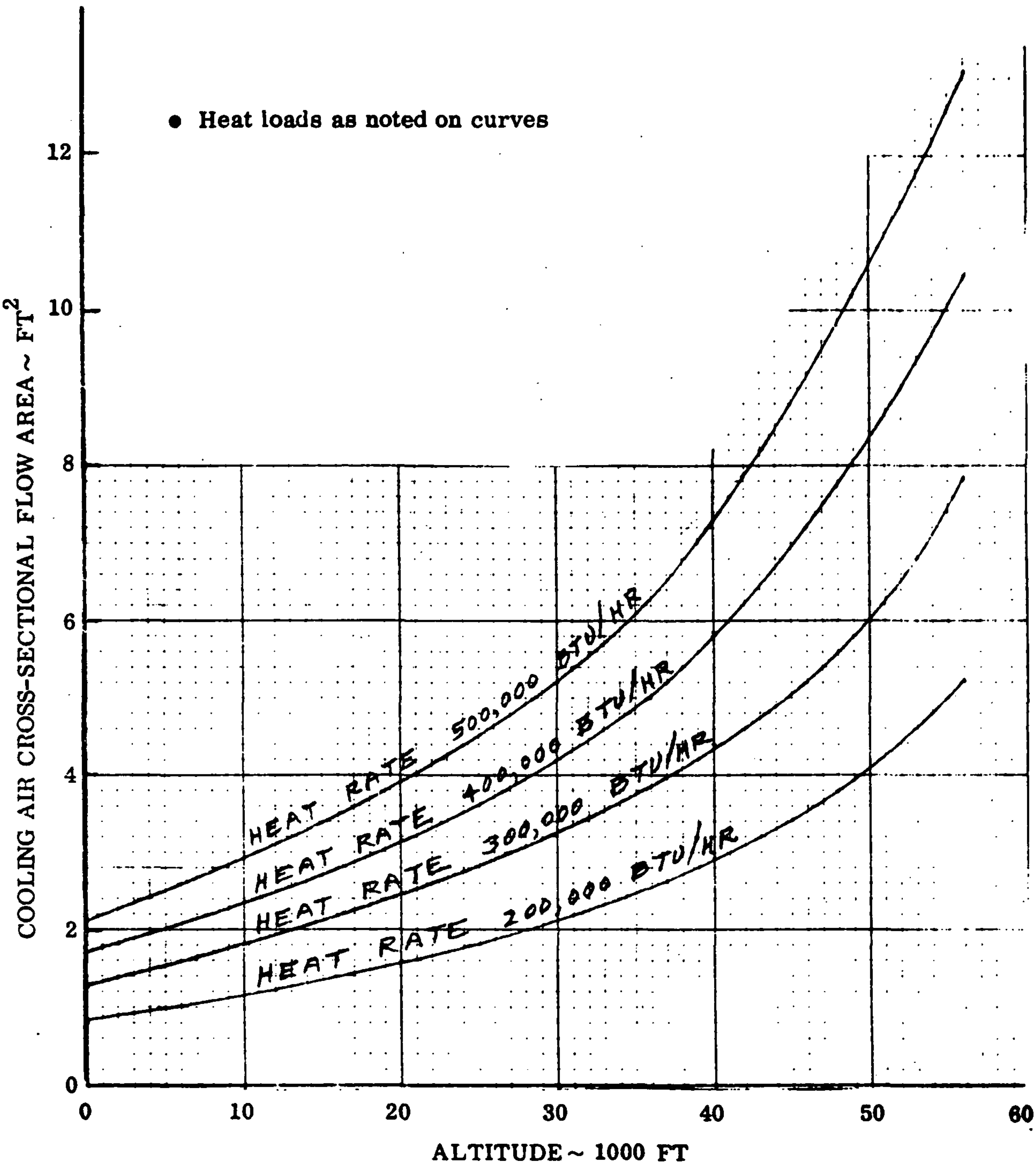


Fig. 3-4 **Reentry Ram Air Cooling Duct and/or Heat Exchanger Cross-Sectional Air Flow Area for M = 0.3 Flow, Based on Air Flow Rate for 10°F Air Temperature Rise**

of 10°F. Figure 3-5 shows the corresponding distance by which the folded panels must be separated to achieve the required flow area, again as a function of altitude and heat exchange rate. Figure 3-6 shows the axial length of the air flow path between the radiators required to accommodate the indicated heat exchange rates with convective coefficient corresponding to M = 0.3, versus altitude. At approximately 13,000 feet the ram air temperature is equal to the desired Freon outlet temperature; hence, the required heat exchange surface area becomes infinite at this altitude, and the axial air flow path length becomes infinite. The analysis assumed no fins on the radiator heat transfer surfaces.

In view of the large area requirements associated with ram air cooling of the Freon in the radiators, due to relatively low Freon-to-air temperature difference, high heat loads, and absence of fin convective effects, and in view of the inability of achieving the desired Freon outlet temperature below about 13,000 feet, alternative use of ram air was sought. The possibility of dissipating the hydraulic heat load in a conventional fin-and-tube, oil-to-air heat exchanger was explored. For purposes of this analysis, a hydraulic heat load of 300,000 Btu/hr was assumed, with the oil being cooled from 200°F to 170°F.

The cooling air pressure available for pressure drop through the heat exchanger was assumed to be 50 percent of the free-stream velocity head, allowing the other 50 percent for inlet recovery, duct, and exit losses. The characteristic curves of a known representative heat exchanger were consulted, which show heat transfer rate as a function of air-oil inlet temperature difference and air flow rate, and air pressure drop versus air flow rate. With the known air and oil characteristics, the adequacy of this known heat exchanger for the hydraulic cooling task was evaluated, and it was found that the equivalent of slightly more than two such heat exchangers would be adequate for the 56,000-foot condition. The resultant estimated fin-and-tube heat exchanger size, established by the 56,000-foot condition, was calculated to have 17 by 17-inch core face dimensions and 3-inch thickness. Figure 3-7 shows the required core frontal area versus altitude.

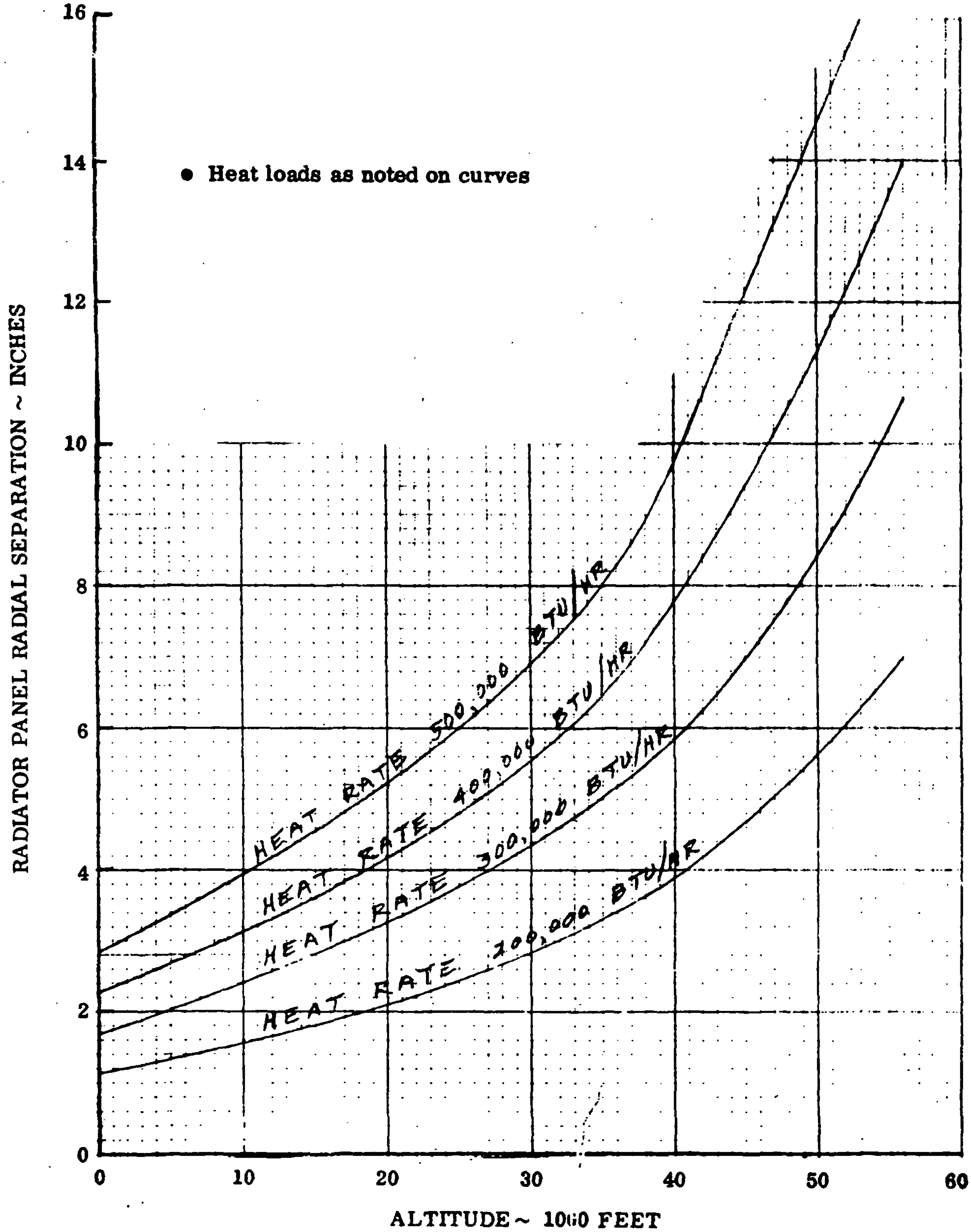


Fig. 3-5 **Distance Between Panels of Stowed Space Radiator for Reentry Ram Air Cooling With M = 0.3 Flow, Based on Air Flow Rate for 10°F, Air Temp Rise**

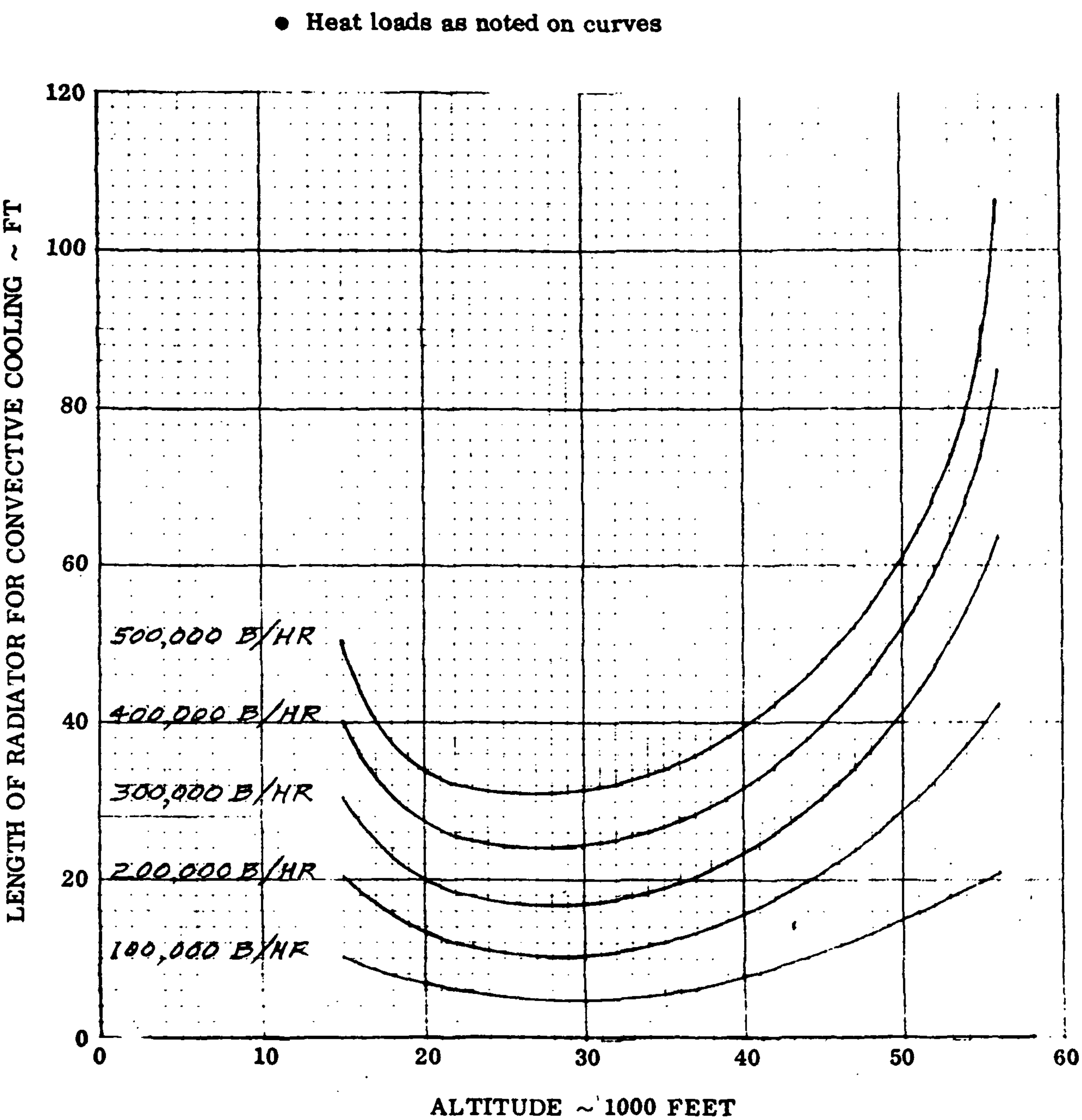


Fig. 3-6 **Length of Space Radiator Air Flow Path for Convective Cooling with Ram Air During Reentry with M = 0.3 Panel Flow Based on Air Flow Rate for 10°F Air Temp Rise**

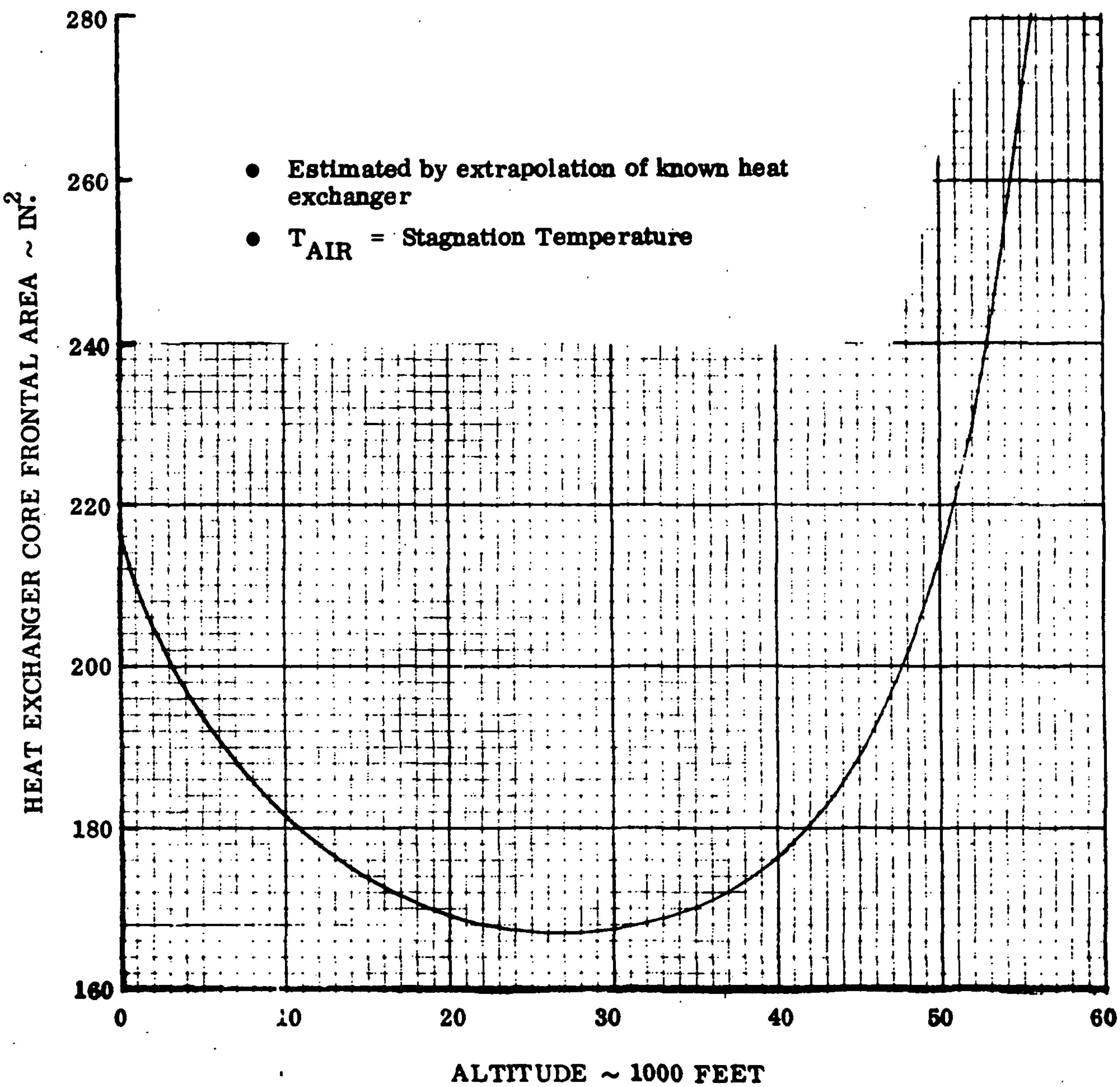


Fig. 3-7 **Required Core Frontal Area of 3-Inch Depth Fin and Tube Heat Exchanger for 300,000 Btu/Hr Hydraulic Heat Load**

General agreement with these size requirements was obtained when checked against a second known heat exchanger of the same type.

3.3.3 APU Comparison Studies

3.3.3.1 Objective. The objective of this study is to compare various APU systems on a weight basis, and to include the function of providing EC/LSS and APU cooling with the comparison. When the Space Shuttle was redirected and the APU system was changed to use hydrazine (N_2H_4) instead of oxygen and hydrogen (O_2 - H_2) as the reactant, a potential heat sink (the cryogenic oxygen and hydrogen) was lost. Prior weight comparisons had indicated that a great weight difference between the N_2H_4 APU and the O_2 - H_2 APU systems did not exist. However, these analyses did not consider the additional cooling capability that the O_2 - H_2 APU system has. Therefore, a new weight study, including the cooling requirements, was performed. Parametric data on a hydrazine APU, a hybrid APU employing both hydrazine and hydrogen, and an oxygen-hydrogen APU (as supplied by the Sundstrand Corp.) were used to conduct the study.

3.3.3.2 Data and Assumptions. It was assumed that three 300 hp APUs would be used and that two of them would be operated at all times. Each one is sized to provide all the power required and therefore only one is required to operate; the other two are used to provide FO-FS capability. The following power profile was assumed for each APU:

Period	Power (hp)	Turbine Discharge Pressure (psia)	Time (Min.)
Prelaunch	87	15	3
Checkout	32	15	12
Boost	32	10	3.5
Coast	32	5	0.2
Insertion	32	5	3.25
Reentry	32	5	50
Reentry	75	5	25
Cruise	147	10	1.25
Cruise	32	10	1.25
Cruise	85	15	1.33
Approach	85	15	.675
Flare	105	15	.225
Touchdown	135	15	.45
Touchdown	32	15	.45
Go-around	150	15	1.25
Go-around	135	15	2.50
Go-around	105	15	1.25

The 32 hp points in the above table are based on the assumption of three hydraulic pumps per APU, each absorbing 8 hp while idling at rated speed, and one alternator per APU, absorbing 8 hp.

Propellant delivery pressure to the APU was assumed to be 500 psia and 500°R for all propellants.

The EC/LSS heat load during reentry and descent was assumed to total 363,000 Btu, released at an average rate of 4,239 Btu per minute during the 85.63 minutes of reentry and descent events shown in the above table.

The APU information was based on data received from Sundstrand Aviation, as shown in Figs. 3-8 to 3-17.

Wt. Includes: Turbine Assembly, Gearbox, Control System, Decomposition Chamber, Turbine Containment, Lube Oil Heat Exchanger, Electrical Controller

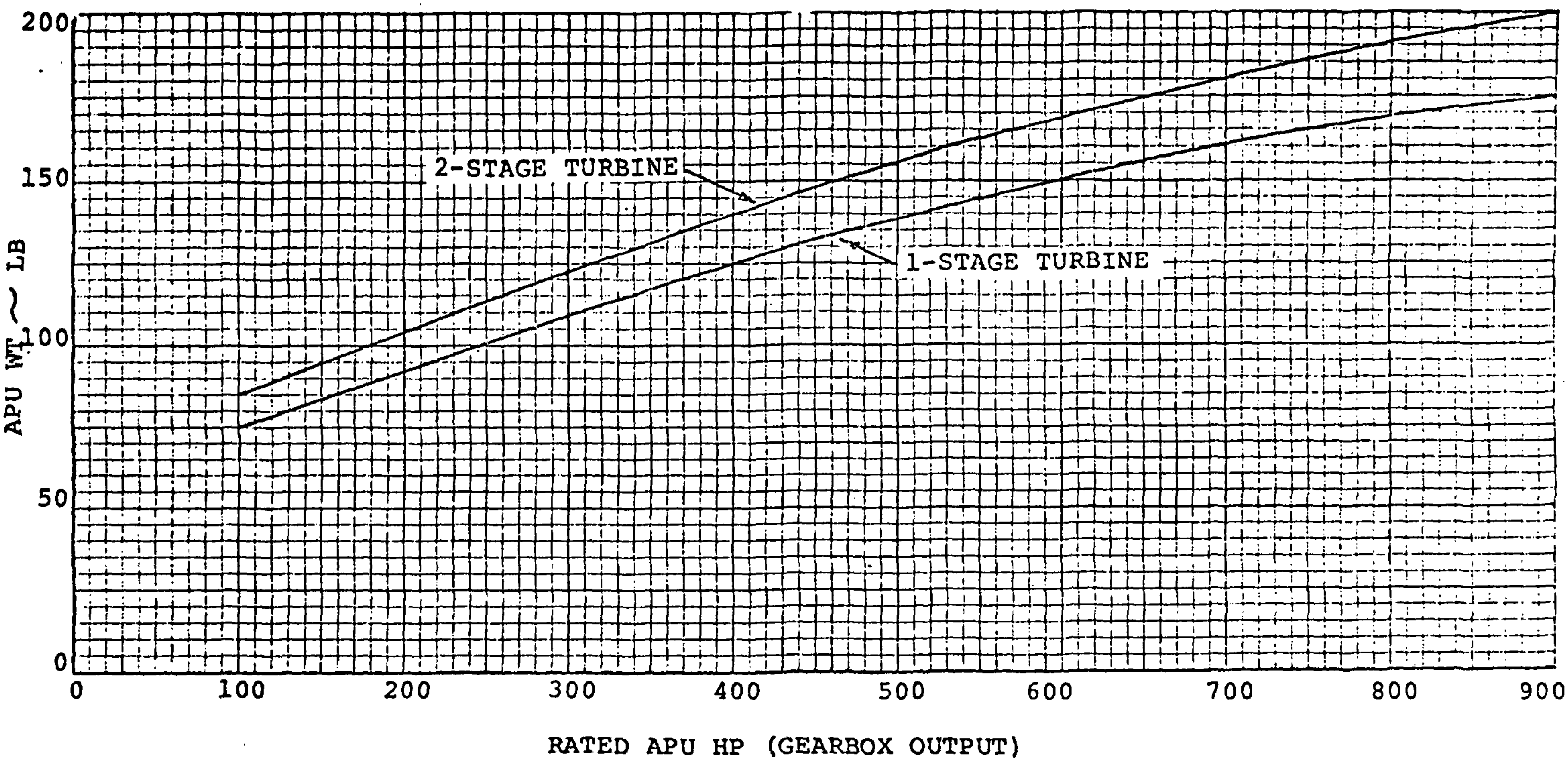


Fig. 3-8 Hydrazine APU Weight

(2-STAGE PRESSURE COMPOUNDED TURBINE)

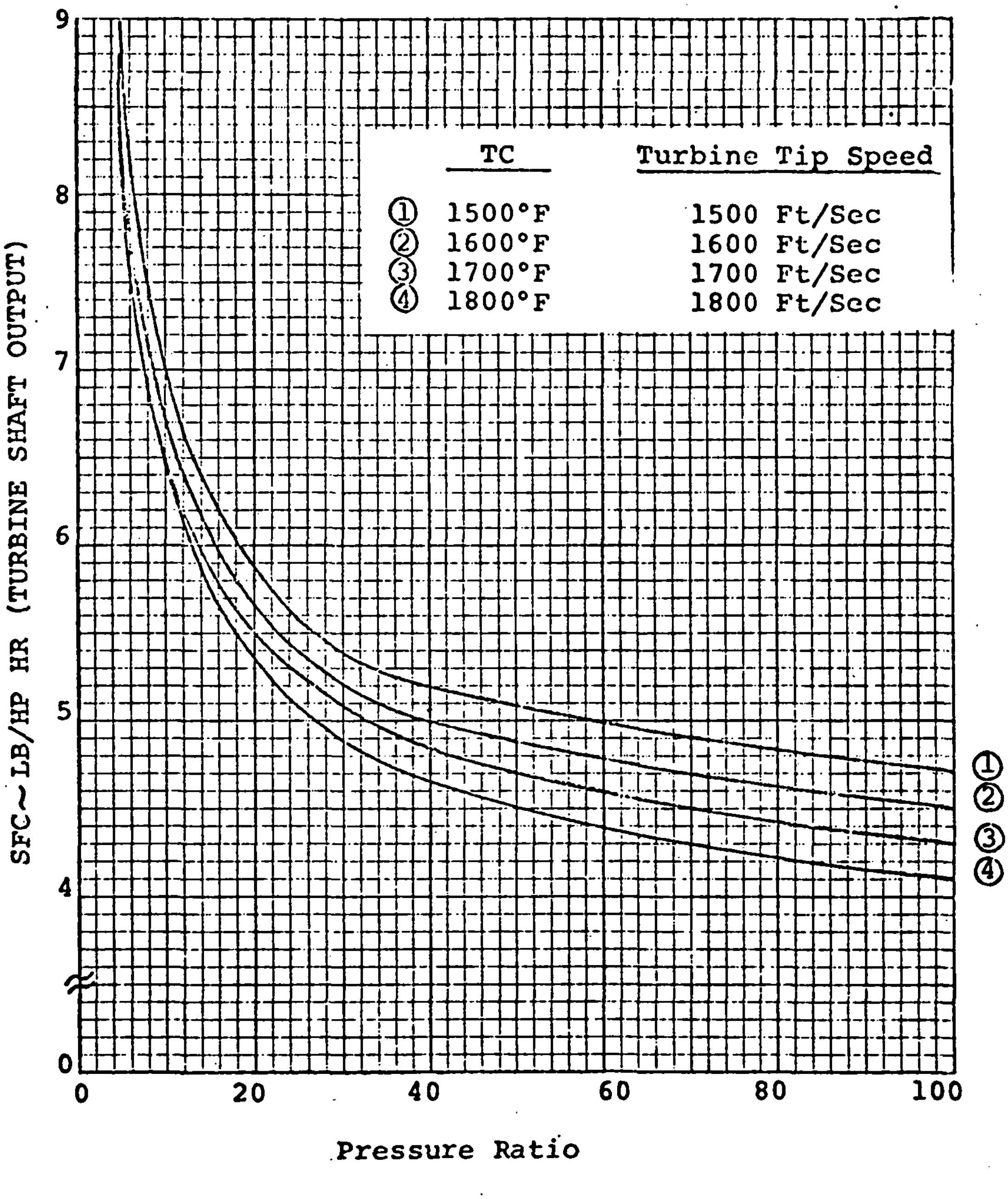


Fig. 3-9 Hydrazine Specific Fuel Consumption

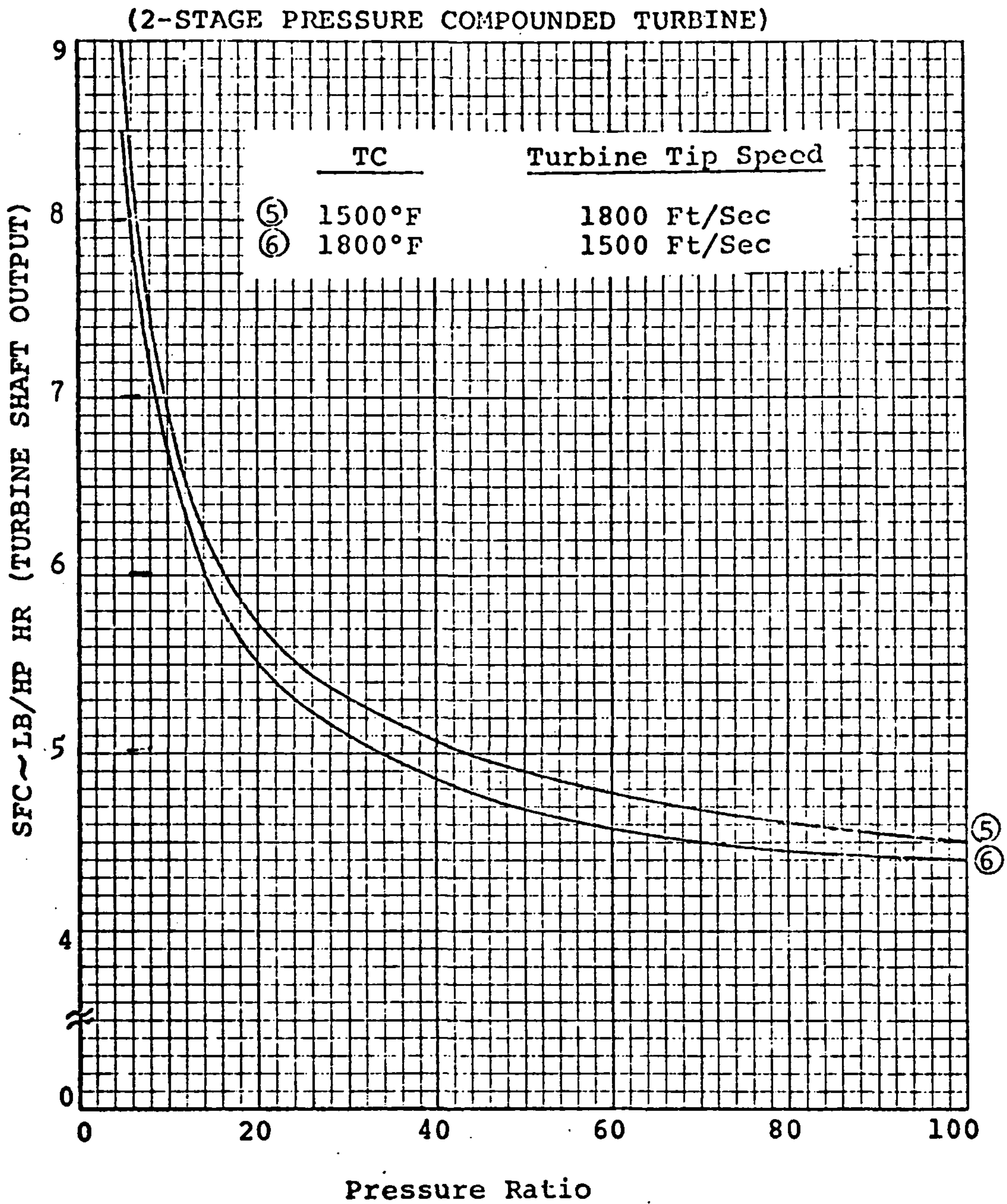


Fig. 3-10 Hydrazine Specific Fuel Consumption

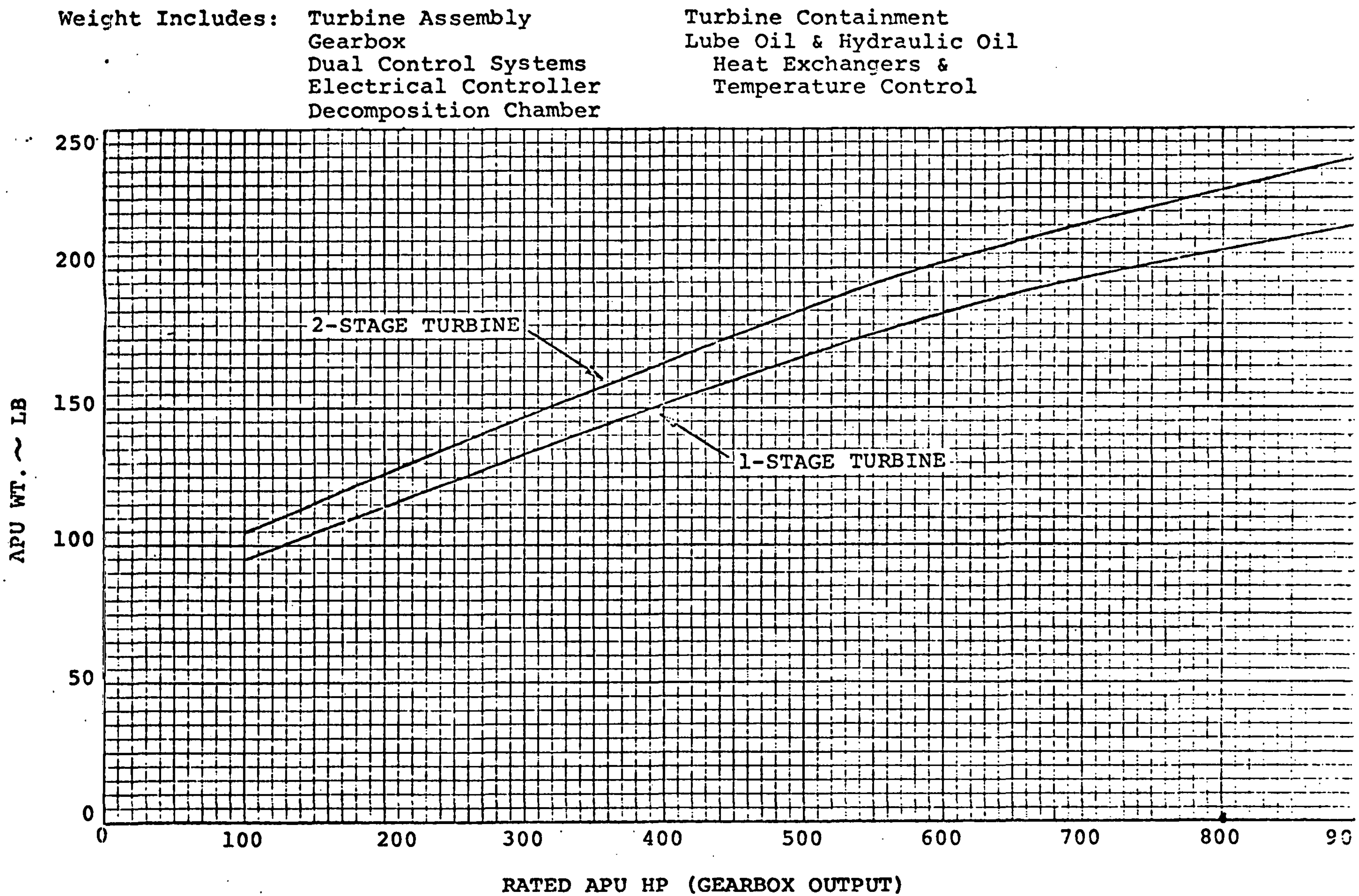


Figure 3-11 Hybrid Hydrogen/Hydrogen APU Weight (Common Turbine)

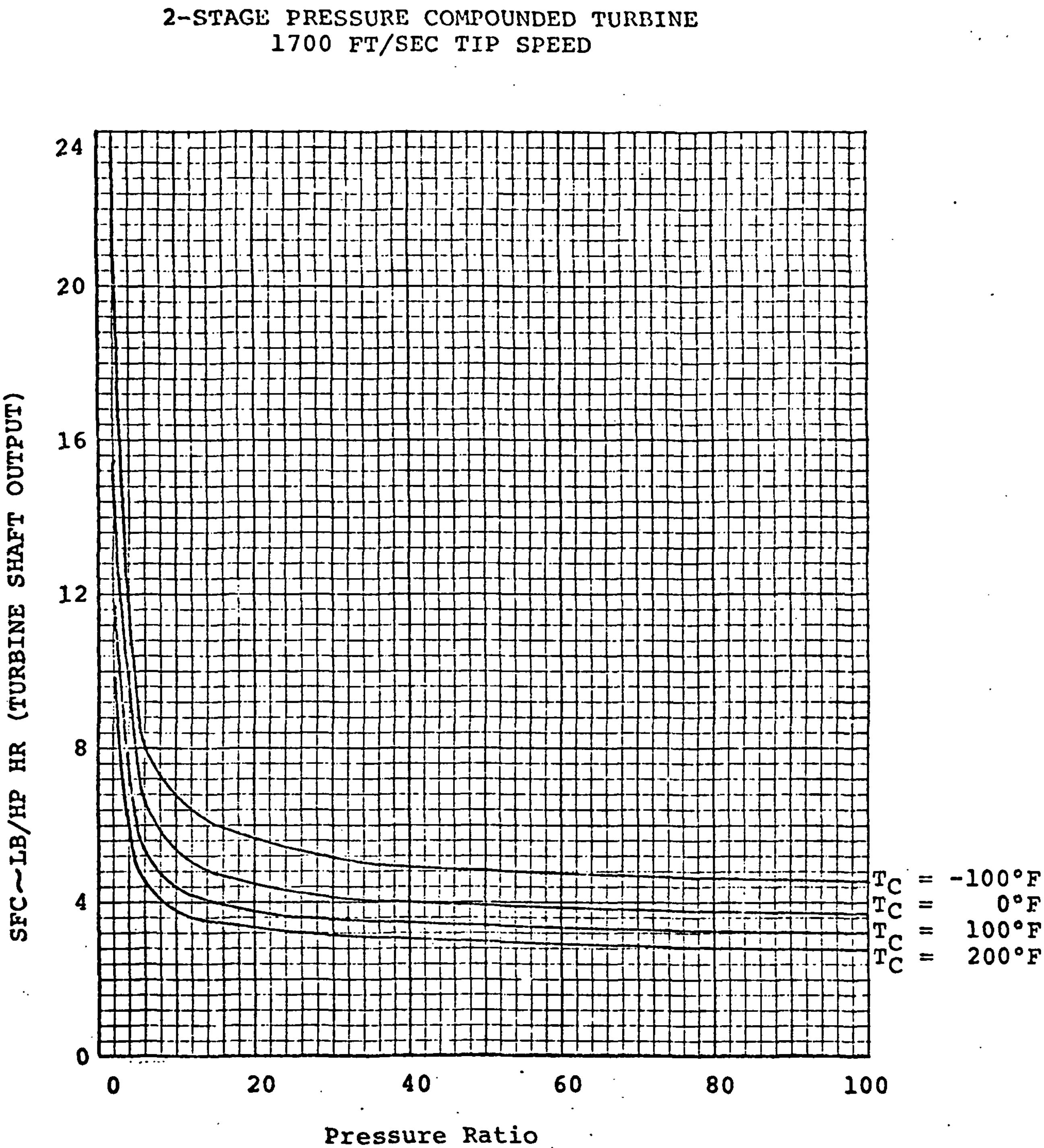


Fig. 3-12 Hydrogen SFC for Hybrid APU

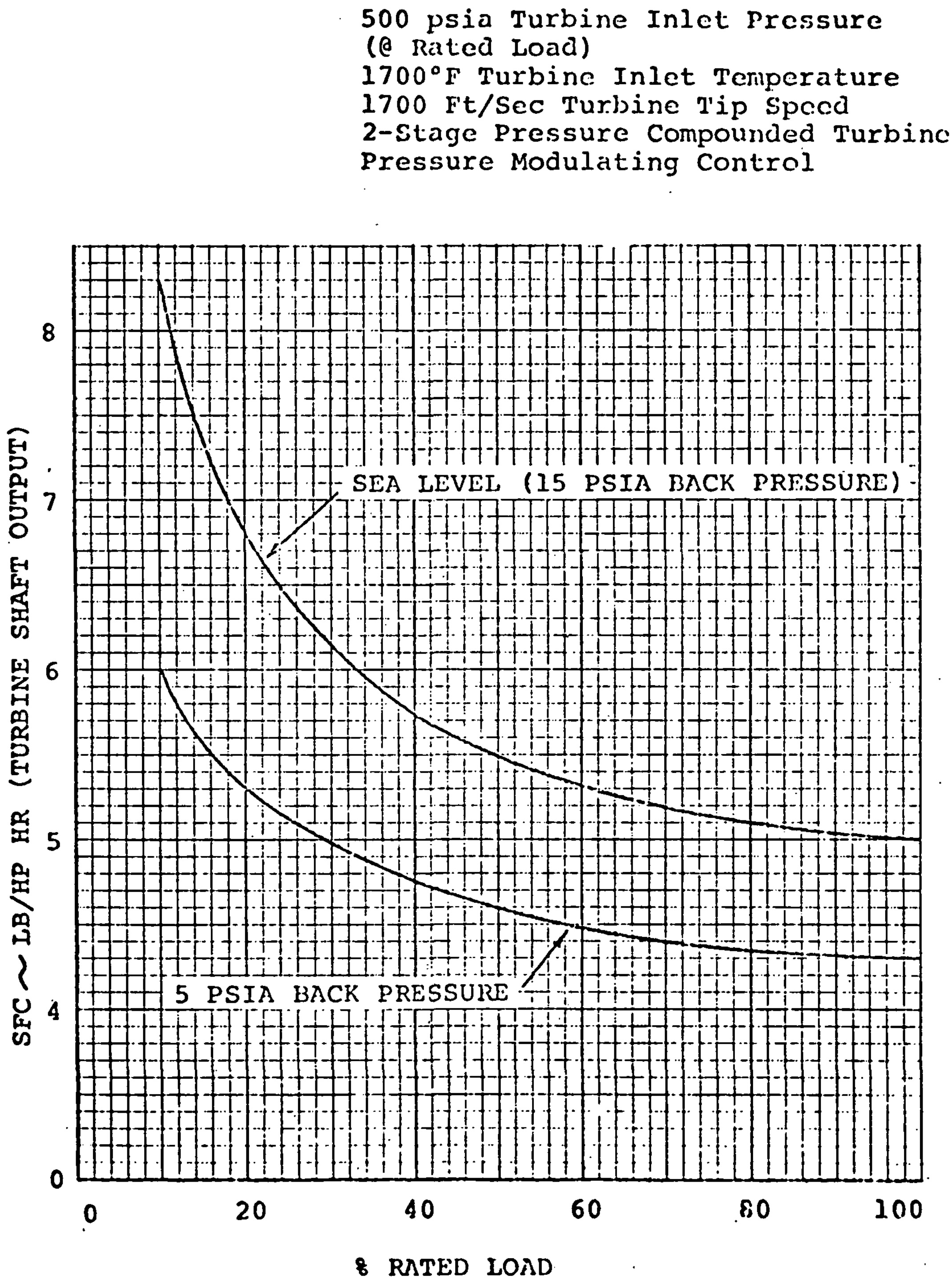


Fig. 3-13 SFC vs Percent Rated Load for Hydrazine APU

$H_2 - O_2$

2-Stage Re-entry Turbine
1500 Ft/Sec. Tip Speed
1500°F Inlet Temp.
1000 Hr. Life

WEIGHTS INCLUDE

Turbine, Gearbox, Combustor, Turbine Tri-Hub Burst Containment, Speed Control, Gearbox Lube Pump

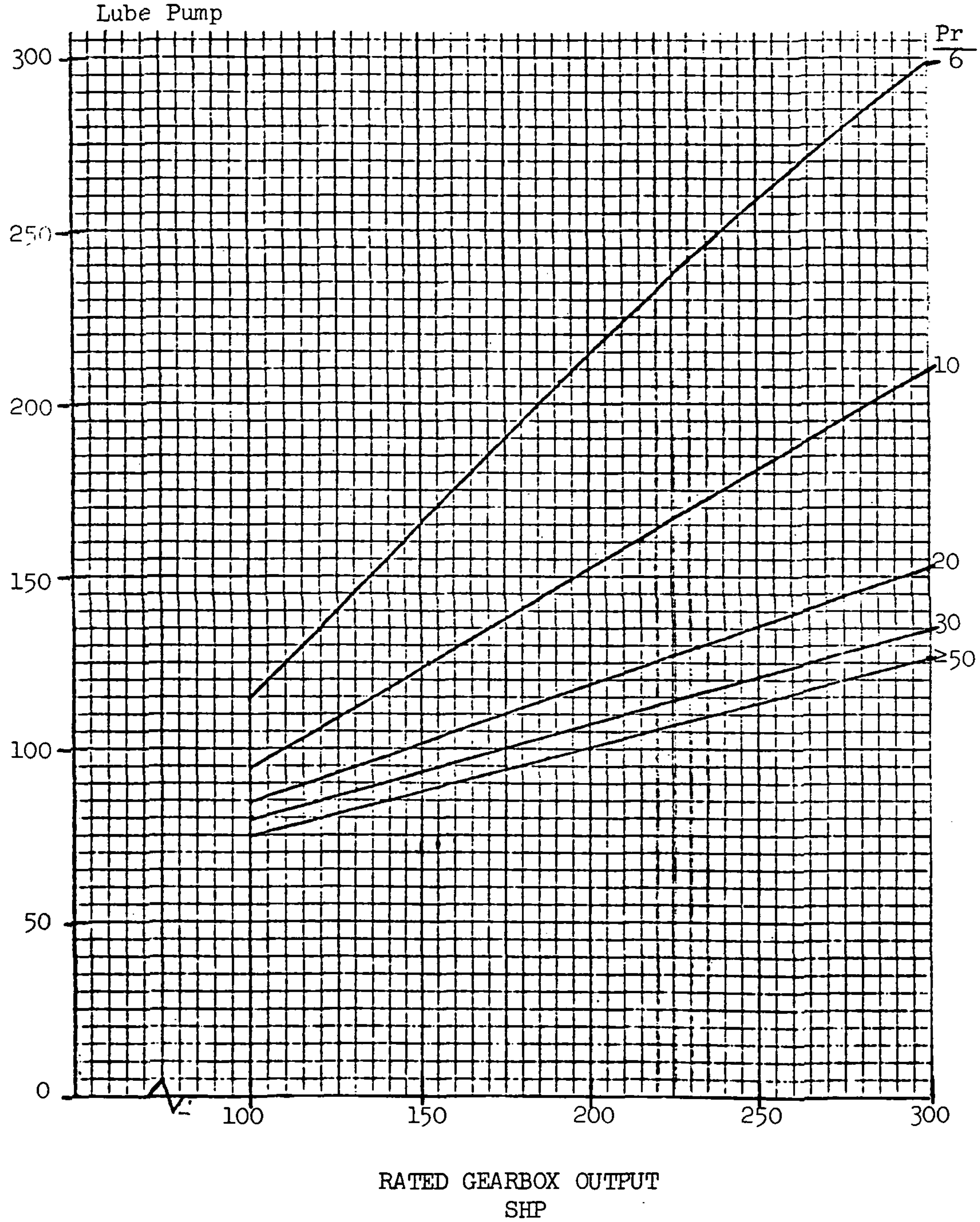


Fig. 3-14 O_2 H_2 APU Weight

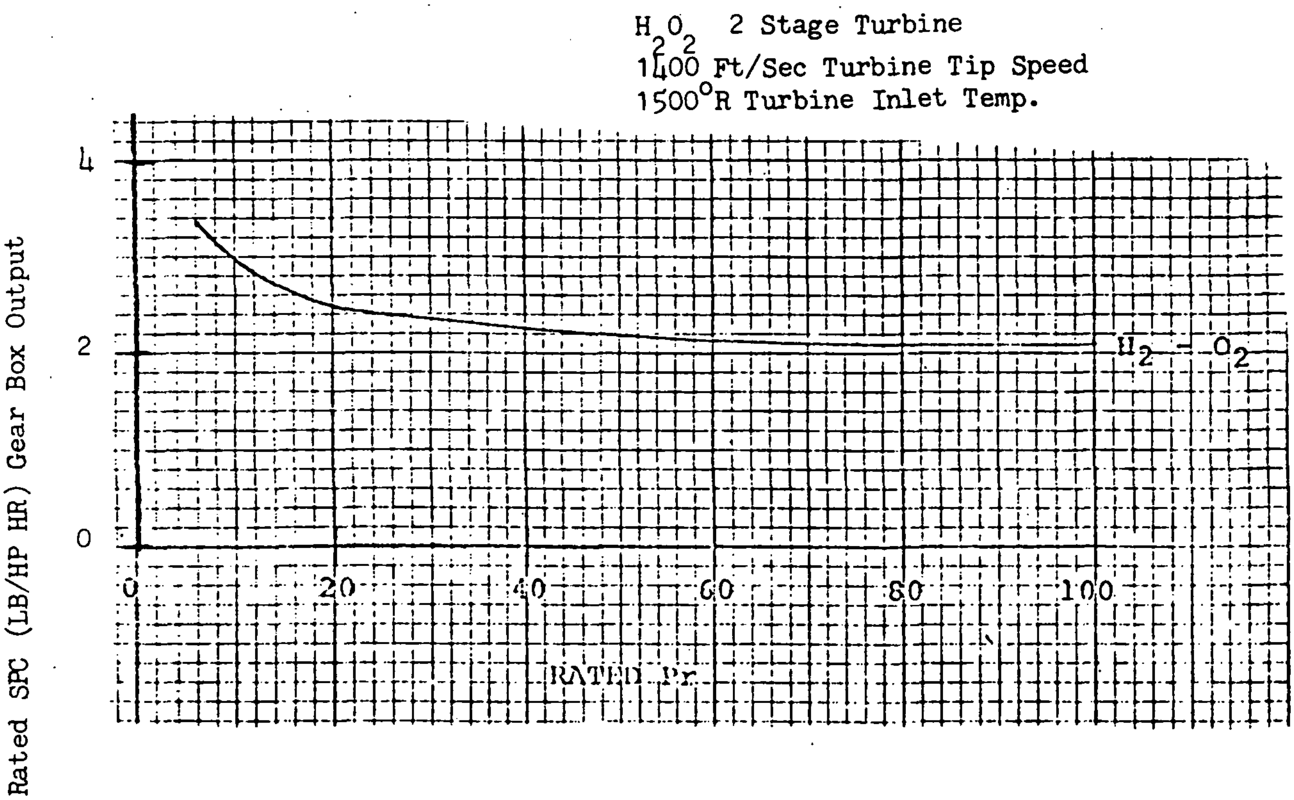


Fig. 3-15 Rated Specific Propellant Consumption Versus Pressure Ratio O_2 H_2 APU

OFF-design SPC
Modulating Propellant Control
.81 O/F Ratio

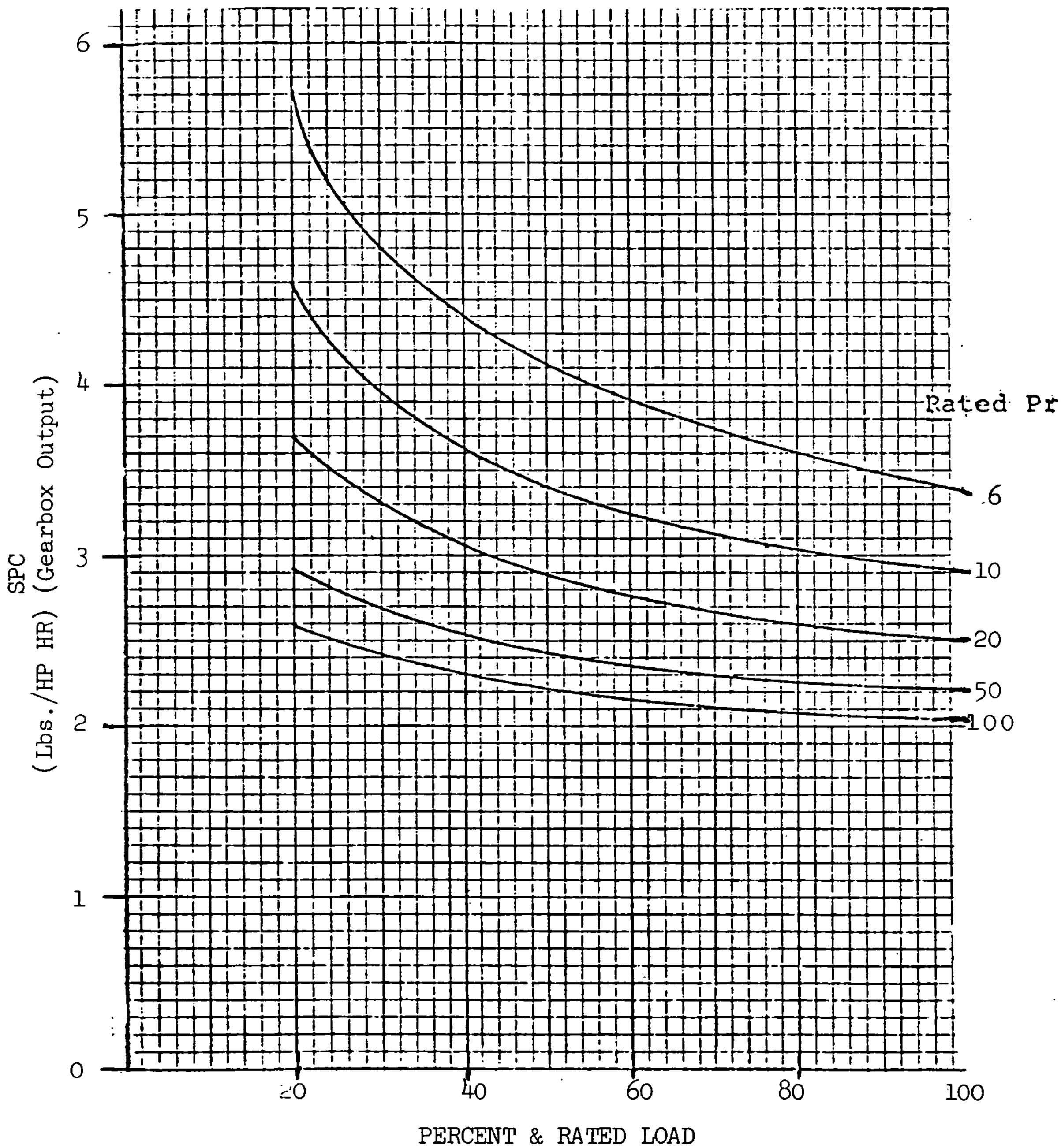


Fig. 3-16 O_2H_0 SPC vs Percent Rated Load

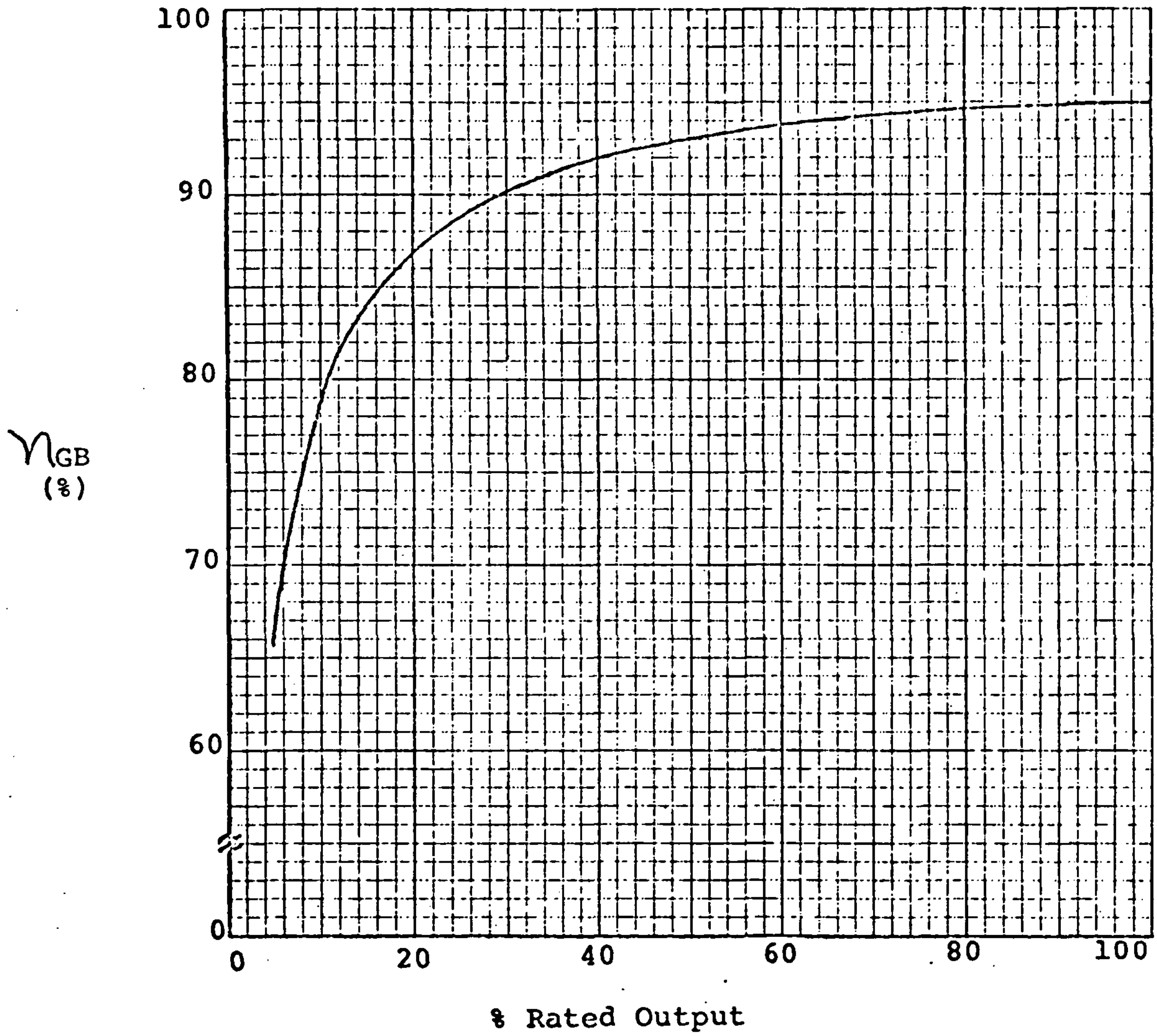


Fig. 3-17 Typical Gear Box Efficiencies

3.3.3.3 Procedure. The procedure for comparison of the alternate propellant systems was as follows:

1. For the N_2H_4 system, for each interval in the power profile, determine the APU gearbox efficiency, turbine work, specific propellant consumption, and total propellant used during the interval. Sum the resultant N_2H_4 quantities for the mission, and determine the sum of system weights associated with that sum. For the reentry and descent portions of the profile, a quantity of dedicated hydrogen and its associated equipment was charged against this sytem for purposes of absorbing the EC/LSS heat load.

2. For the hybrid N_2H_4/H_2 system, the first 32 hp of power requirements during reentry and descent was assumed to be met by H_2 propellant flow to the hybrid APU, and all power increments above 32 hp were assumed to be met by use of N_2H_4. The H_2 was assumed to be heated to 500°R, by the EC/LSS heat load. Under these conditions, the required quantities of H_2 and N_2H_4 were determined, and the weights of system elements associated with these quantities were determined.

3. For the H_2-O_2 system, the same procedure was followed as for the N_2H_4 system. The H_2 was assumed to be heated to 500°R by the EC/LSS heat load and/or APU exhaust, and the O_2 was assumed to be heated by the APU exhaust. Each interval in the power profile was investigated to determine whether the H_2 flow would be adequate to match the average EC/LSS heat load of 4239 Btu per minute. For the intervals where the H_2 flow was inadequate (two only), dedicated H_2 was assumed to be used to make up the difference, and was charged against the system.

3.3.3.4 Discussion. Figure 3-18 shows the heat absorption capability of hydrogen, based on a temperature rise from 50°R to 500°R at 500 psia. This curve was referred to for determination of the H_2 propellant and dedicated H_2 heat capacities.

Table 3-3 shows a comparison of system weights for the three types of propellant systems. Hydraulic components and the EC/LSS-hydrogen heat exchangers are omitted because of their commonality to the three systems. In each of the three cases, the APU weights are based upon a two-stage turbine, and the propellant weights reflect two-stage efficiencies. For the hybrid N_2H_4/H_2 system, the N_2H_4 and H_2 are considered to operate on the same turbine wheels, with partial-arc admission for each gas.

All tanks were calculated for 0.025-inch minimum gage aluminum wall thickness, since this resulted in higher weights than those calculated by assuming a safety factor of 1.35.

The H_2-O_2 system shows a weight advantage over the other two systems.

3.3.4 Cryhocycle Description

3.3.4.1 Introduction. One of the ways to balance the heat being generated with the available cooling is to use gas expansion machines coupled to electrical power generators instead of full cells. The fuel cells must reject heat at about 2100 to 2700 Btu/kWhr depending on module size and design. This heat is usually rejected via full cell module heat exchangers, coolant loops, and space radiators. If this source of heat can be eliminated, then the radiator size can be reduced. The Cryhocycle is an excellent way of achieving this.

The Cryhocycle is an expander-generator system that can use liquid hydrogen plus ambient temperature heat to produce electrical power and a net cooling effect. These characteristics appear to be well suited to the needs of the Space Shuttle and a closer inspection of the Cryhocycle is thus appropriate. To perform a preliminary assessment of the Cryhocycle for the Space Shuttle,

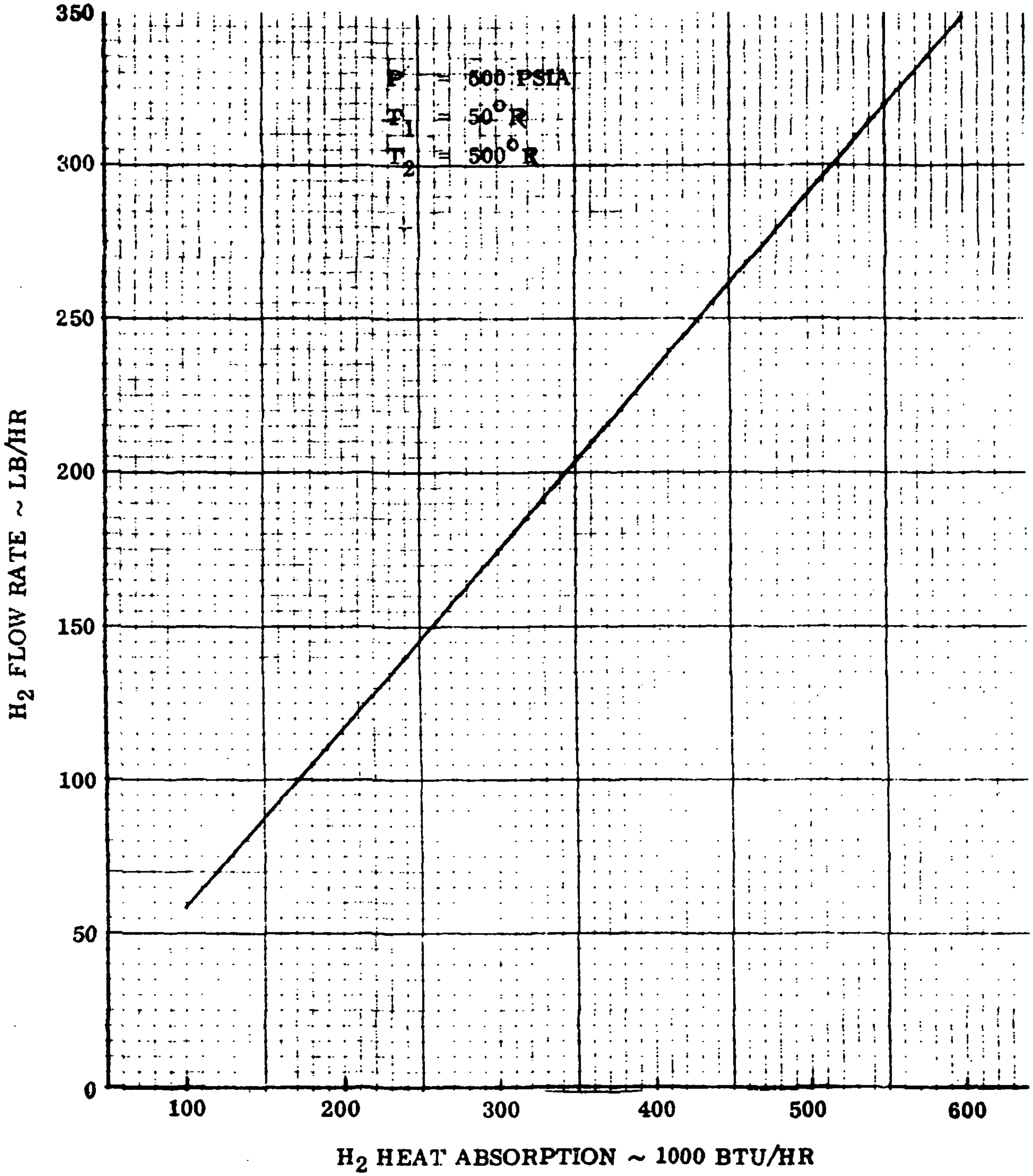


Fig. 3-18 H_2 Heat Absorption Rate Vs H_2 Flow Rate

C-2.

Table 3-3

SUMMATION OF APU SYSTEM WEIGHTS (LB)

	N_2H_4	Hybrid N_2H_4/H_2	H_2 - O_2
3 APUs(1)	366	440	390
N_2H_4 Propellant	1206	610	-
H_2 Propellant	-	234	277
O_2 Propellant	-	-	224
Dedicated H_2	191	0	25
3 Alternators(2)	30	30	30
Installation(3)	217	217	217
N_2H_4 Tank	21	16	-
H_2 Tank System	78	123	133
O_2 Tank System	-	-	24
Pressurization	12	29	27
3 N_2H_4 Pumps	6	5	-
3 H_2 Pumps	-	19	21
3 O_2 Pumps	-	-	3
Totals	2127	1723	1371

(1) Includes turbine assembly, gearbox, control systems, decomposition chamber, turbine containment, oil heat exchanger, and electrical controller

(2) 10 KVA each

(3) Includes sump oil weight, ducting mounting structure, and oil cooling piping for three APUs

certain parametric data are required, relating to the power output, cooling capacity, liquid-hydrogen consumption, machinery weight, reliability, and operating characteristics. At the preliminary design stage, these data will necessarily be approximate and will serve mainly to indicate the general possiblities of the system, and to provide a basis for determining whether the Cryhocycle offers enough system advantages to warrant further work.

Sundstrand Aviation has examined the Cryhocycle system and has issued a textbook that contains a detailed analysis of the device and includes most of the parametric data necessary for preliminary design. This textbook is, in our opinion, of very high technical quality and is most comprehensive. The data therein have been condensed, rearranged, or augmented as necessary to provide the type of information desired for this study.

The following aspects of the Cryhocycle are described in subsequent sections.

1. A description of how the Cryhocycle works.
2. A summary of the basic operating parameter choices, and their relative influence in cycle efficiency.
3. Obtainable ratios of net cooling capacity to electrical power generation.
4. Machinery weights.

In the following, a distinction is made between heat generated as a result of dissipation of electrical energy and heat arising from sources such as metabolism or aerodynamic heating. The heat arising from electrical dissipation will be exactly equal to the electrical power produced by the Cryhocycle. This heat is fed back to the Cryhocycle almost 100 percent in a process which is basic to the operation of the system. The other sources of heat are referred to as external or non-electrical sources. The Cryhocycle requires a certain minimum amount of external heat to operate, but can accept up to a certain maximum quantity.

3.3.4.2 Basic Principle. The Cryhocycle is basically a process for using stored liquid hydrogen to produce electrical power with the addition of very little net heat to the system. One version of the Cryhocycle is shown in Fig. 3-19 and 3-20. The thermodynamic process, component schematic, and description of operation have been selected to illustrate the essential components and systen considerations. An actual system would be more complex and less easy to describe, but would show the same basic process phases. Liquid hydrogen is stored subcritically at state 1. A flow of hydrogen is withdrawn from the storage vessel and is compressed to a substantially higher pressure, point 2. It is then warmed to point 3a in a counter flow heat exchanger. Because of the inefficiency of this regenerative heat exchanger, plus non-ideal gas behavior, the outlet temperature of the high-pressure hydrogen will be less than that of the warm side of the exchanger. This temperature deficiency is made up by heating the hydrogen from point 3a to 3b in a make-up heat exchanger. The heat supply to this exchanger is from a non-electrical source.

The hydrogen flow then enters the electrical power generation system. The flow is warmed to a point 3 by contact with a circulating fluid which conveys heat from the electrical power load back to the working fluid. The gas passes through a control throttling valve, points 3 to 4, and is then expanded to point 5 in a recriprocating, or turbine, expander. To increase the efficiency, several stages of expansion may be used to cover the available overall pressure ratio. In the three-stage process shown, the hydrogen is rewarmed by the electrical load coolant, points 5 to 6, expanded from point 6 to 7, rewarmed, point 7 to 8, and expanded to the system low pressure at point 9. The electrical load coolant loop passes over the generator gear box and expander bearings as well as the electrical load and thus picks up from one or other source all the energy output of the expansion process. The hydrogen enthalpy at point 9 will be the same at 3b and thus the system of electrical load and expander can be considered adiabatic. This situation holds strictly only if all the power produced by the expansion process is dissipated at ambient temperature. In fact, a certain portion of the electrical energy will be dissipated outside the adiabatic region at a

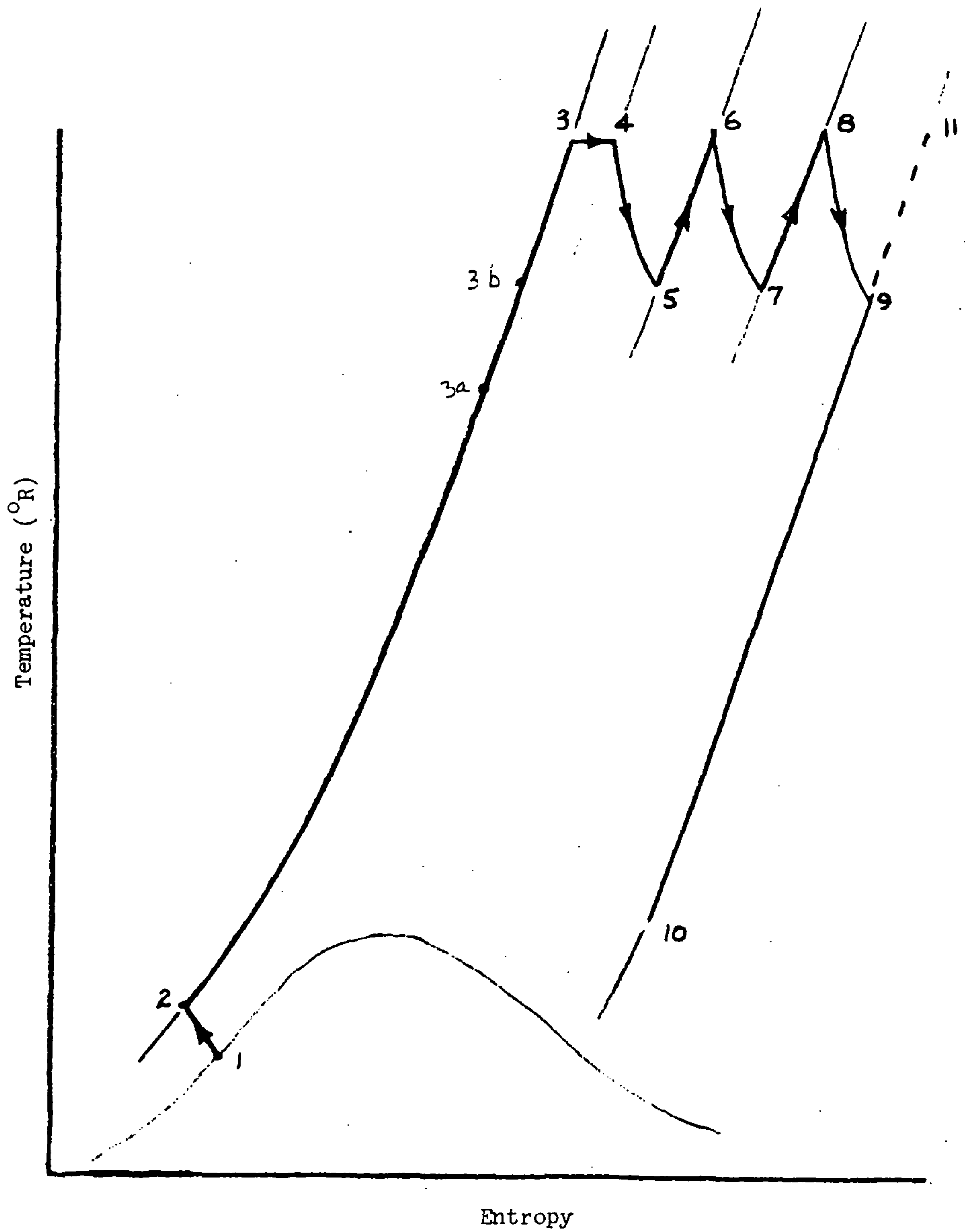


Fig. 3-19 The Basic Cryhocycle Process

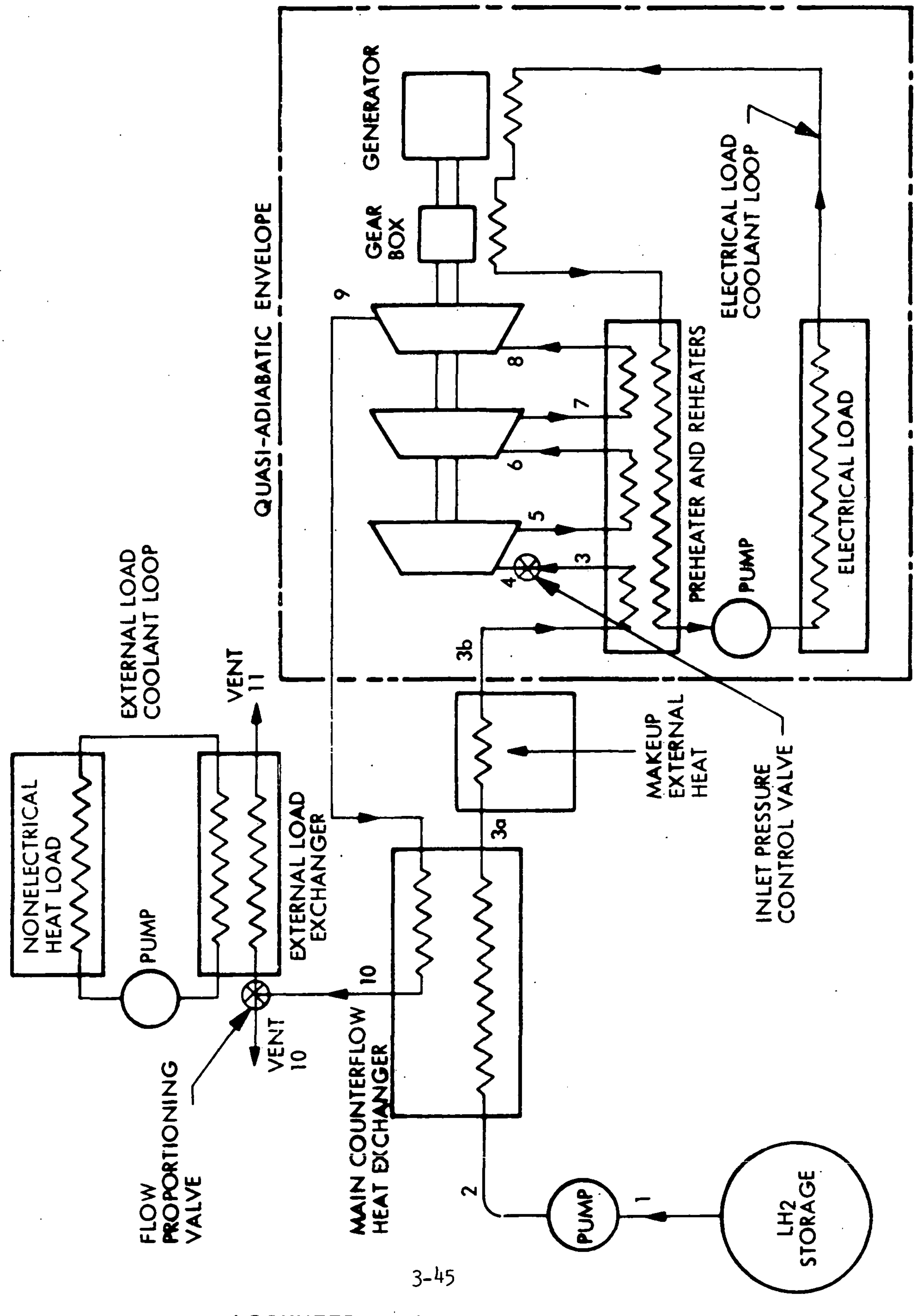


Fig. 3-20 Basic Cryhocycle Component Schematic

temperature lower than ambient. For example, the energy imparted to the hydrogen by the low-temperature pump will not be reclaimable at ambient temperature. This will mean that the enthapy of the hydrogen at point 9 will be lower than at 3b. This will require additional heating of the high-pressure hydrogen in the make-up heat exchanger, over and above that required to offset the effect of heat exchanger inefficiency and non-ideal gas behavior. The low-pressure hydrogen gas at point 9 is passed through the warm side of the regenerative heat exchanger, warming the high-pressure flow in the process, and is then discharged at point 10. The system described is thus able to produce electrical power in a process which is virtually self-sustaining because of the energy feedback from the electrical load to the preheater and reheaters. The feedback process is not 100 percent effective, however, and a small supply of non-electrical heating is required. This can be drawn from sources such as metabolic or aerodynamic sources, but in the interest of making the process entirely self-sustaining, it is recommended that a supply of oxygen be included with the system. The make-up heat can then be produced in the form of an oxygen/hydrogen reaction.

The Cryhocycle can also be used to produce cooling of non-electrical heat loads. As shown in Fig. 3-20, the low-pressure hydrogen is discharged from the main regenerative heat exchanger at point 10. A portion of this cold gas stream can be diverted through a heat exchanger to absorb heat from the non-electrical heat load coolant. The maximum amount of cooling available will be when the entire flow is passed through this exchanger and is discharged at point 11. In summary, the Cryhocycle is therefore an electrical power-producing device which can absorb an amount of non-electrical heat ranging from a minimum value equal to the requirements of the make-up heat exchanger up to a maximum value equal to the heat required to raise the hydrogen from the discharge temperature, point 10, to the ambient temperature, point 11.

3.3.4.3 Influence of System Parameters on Cryhocycle Efficiency. The efficiency of the Cryhocycle can be measured by its specific hydrogen consumption (SHC), measured in lb/KWhr net available electricity. For the purposes of discussion, the components of the Cryhocycle can be considered

in two sections - those inside the dashed line in Fig. 3-20 and those outside. It will be assumed that sufficient make-up heat will be supplied to keep temperature point 3b equal to temperature point 9. In this case, SHC will depend upon the following parameters.

(a) Overall expansion pressure ratio from 3b to 9

(b) Number of expansion stages

(c) Maximum cycle temperature, the temperature at point 3

(d) Specific heat ratio of hydrogen

(e) Molecular weight of hydrogen

(f) Thermodynamic efficiency of the expander

(g) Mechanical efficiency of the expander

(h) Mechanical efficiency of the gearbox

(i) Mechanical-to-electrical energy conversion efficiency of the generator

(j) Energy required by the low-temperature pump

Hydrogen properties (d) and (e) are, of course, fixed. Variations in the efficiencies, (g), (h), and (i) will be relatively small. Variations in the thermodynamic efficiency, parameter (f), will tend to be small within each basic expander type, but there will be a significantly lower value for the turbo-expander than for the reciprocator. Because of the much higher mass velocities in the turbine, the working fluid can absorb less heat during the expansion process. The expansion thus tends to be closer to adiabatic than in the case of the reciprocator, and the departure from ideal isothermal expansion is greater. The higher mass velocities of the turbine do lead to a smaller expander, however. The pump power will be much less than the gross electrical output of the system and will be a function of pressure ratio, temperature level, and pump efficiency. Of these variables, only the pressure ratio may be varied significantly. The maximum cycle temperature will be the operating temperature of the electrical load and is thus relatively

pre-determined. The major system parameters open to choice are thus pressure ratio and number of expansion stages. The effect of these parameters is shown in Fig. 3-21 for a system with adiabatic expansion, total conversion of expansion work to electrical energy, and zero pump power. It can be seen that to obtain the benefits of multi-staging without introducing undue complexity, between 2 and 4 expansion stages seem reasonable. Also, above a pressure ratio of about 50, little further reduction in SHC can be expected.

3.3.4.4 Recirculation. When little non-electrical heat needs to be removed, the hydrogen working fluid is rejected to space at a very low temperature. This gas could be used as a low-temperature heat sink and thus one could consider adding in parallel, with the open system a closed-cycle power unit which receives heat at ambient temperature and rejects it to the venting low-temperature open-cycle working fluid. Such a system could be entirely self-contained, using its own hardware, and operating at separately optimized conditions. Alternatively, it could be incorporated as a modification to the open system, in which form a minimum of additional hardware would be required, but some compromising of operating conditions would be incurred. It is this latter alternative which has been considered most appropriate by Sundstrand, and which is shown in Fig. 3-22 and 3-23. The result is a closed-cycle system superimposed upon an open-cycle system. The open-cycle system operation is the same as previously described and shown in Figs. 3-19 and 3-20, with the exception that the venting working fluid does not pass through the full length of the regenerative heat exchanger and is thus not fully cooled. Instead, it leaves the exchanger at some intermediate point whose location is determined by the closed-cycle system. The closed-cycle follows point 5-5a-5b-2a-3a-3b-3-4-5. After leaving the first expansion stage, the flow stream is divided into two portions. The open-cycle flow passes through the reheater 5-6 as before; the closed-cycle flow enters the regenerative heat exchanger and is cooled to 5a. There are three passages in this upper section of the exchanger to accommodate processes 2a-3a, 5-5a and 9-10. In the lower section of the exchanger, the closed-cycle flow is further cooled to 5b by the open-cycle flow 2-2a. From 5b to 2a, the closed-cycle flow is compressed to the open-

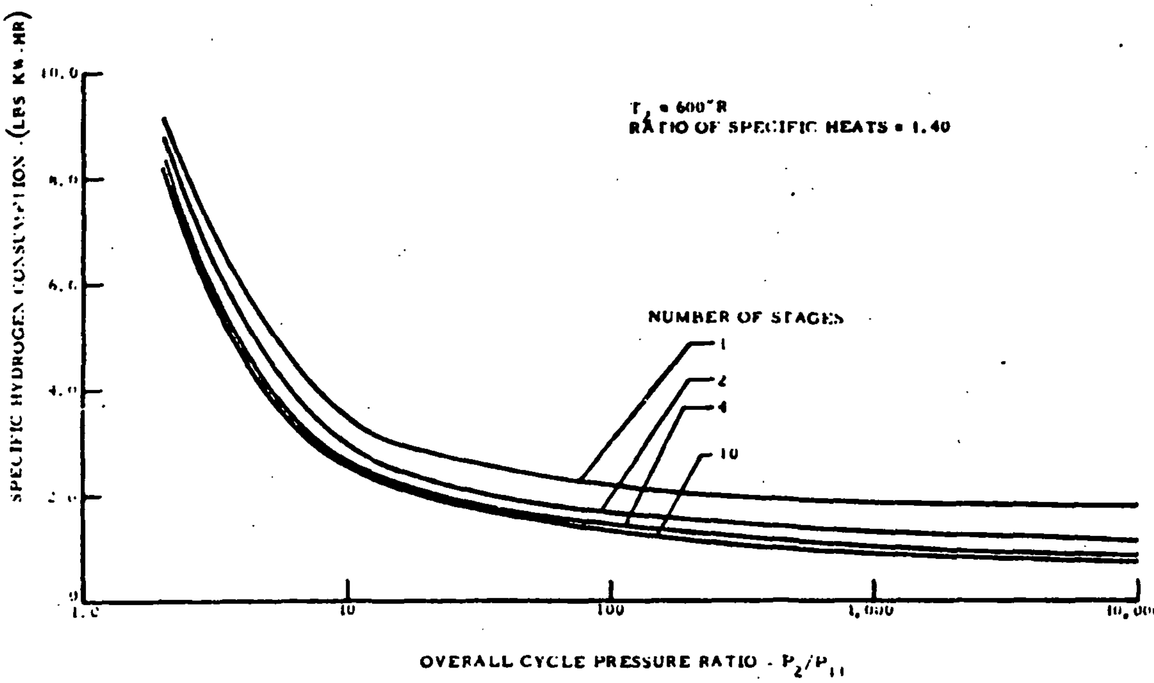


Fig. 3-21 Variation of Specific Hydrogen Consumption with Number of Expansion Stages

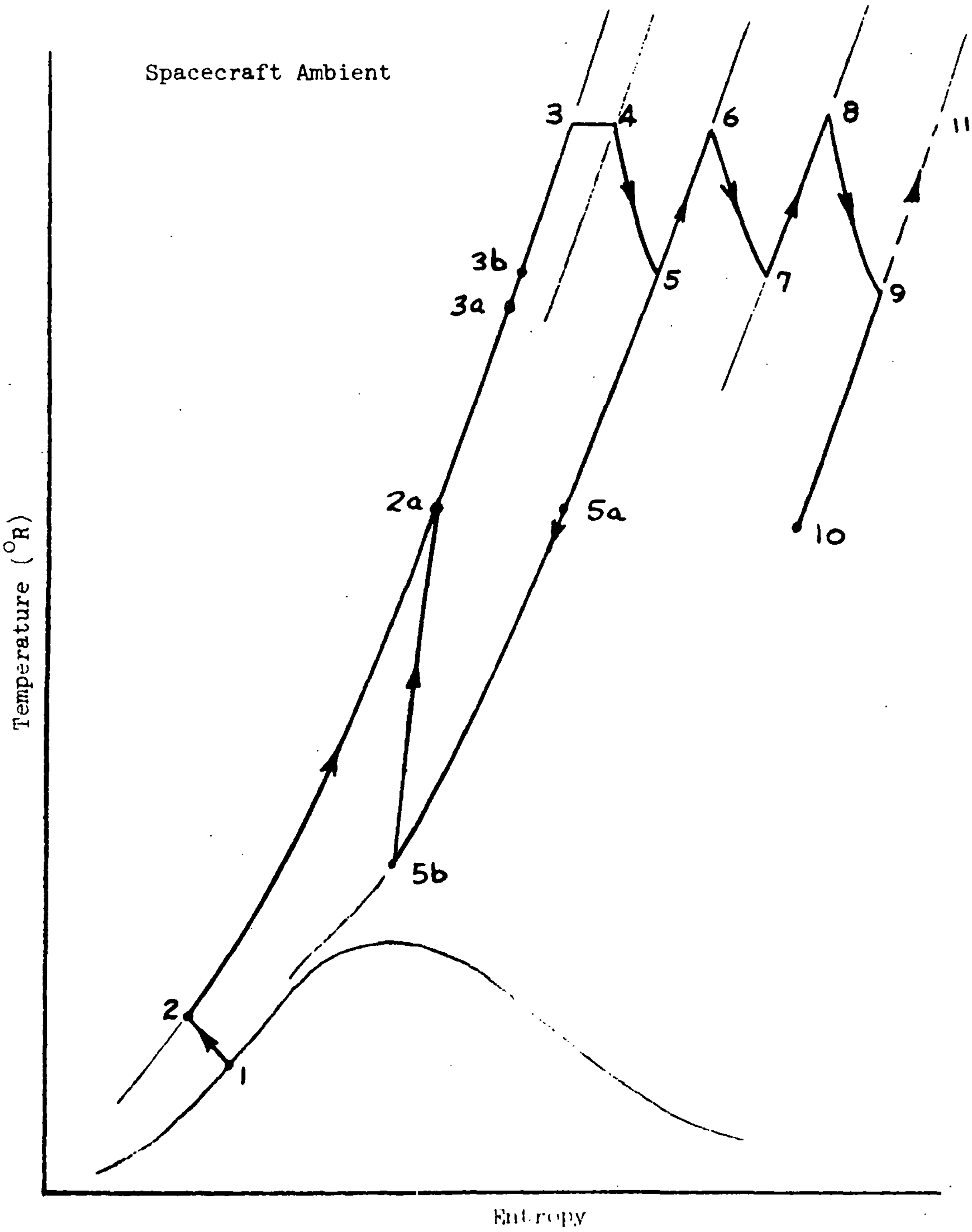


Fig. 3-22 The Cryhocycle Process with Recirculation

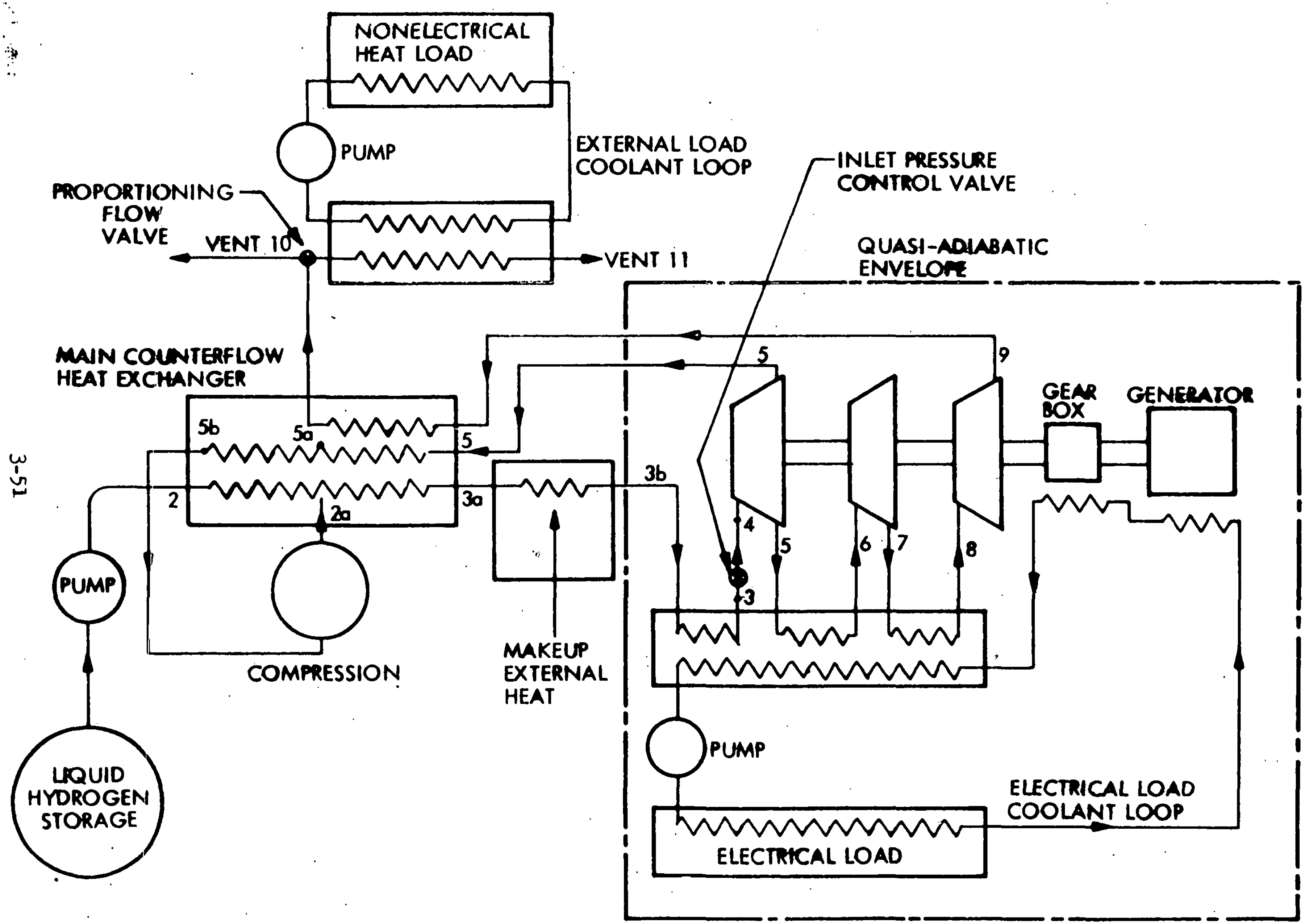


Fig. 3-23 Component Schematic For Cryhocycle With Recirculation

cycle high-pressure valve. The two flows are united at 2a and are warmed in the regenerative exchanger, make-up heat exchanger, and preheater to point 3, where the combined flows enter the expanders. This closed-cycle addition to the Cryhocycle is referred to as recirculation.

When a recirculation loop is added to the open-cycle system, certain additional system parameters must be chosen. They are:

(a) Ratio of recirculation loop flow to open-cycle flow

(b) Number of expansion stages included in the recirculation loop

(c) Compressor inlet temperature (point 5b)

It is shown, in Sundstrand's Textbook, that for a given open-cycle overall pressure ratio there are specific values of flow ratio and compressor inlet temperature which give minimum SHC. The number of expansion stages referes to how many of the open-cycle expansion stages are shared by the recirculation loop. A single-stage of compression may be assumed. Figure 3-24 shows SHC as a function of overall cycle pressure ratio for conditions of zero, one and two stages of recirculation, again for conditions of adiabatic expansion, 100-percent energy-conversion efficiency and zero pumping power for the open cycle. It can be seen that use of a single stage of recirculation results in a substantial reduction in SHC. Additional recirculation stages result in a very small further reduction in SHC, but would require a much larger compressor and larger expander stages because of the higher mass-flow rates. One recirculation stage would thus appear to be about optimum. Recirculation reduces the SHC substantially, but it also reduces the range of non-electrical cooling loads that can be handled. The temperature at point 10 is much higher with recirculation than without. The maximum non-electrical cooling possible is the heat required to raise the hydrogen temperature from point 10 to point 11 and is thus much less with recirculation. Also, a heat balance over the entire process indicates that the non-electrical heat required to sustain the cycle per lb of hydrogen is the difference in the hydrogen enthalpies at points 10 and 1. This quantity is much higher with recirculation. The effect of adding a recirculating loop is thus to lower the SHC to nearly

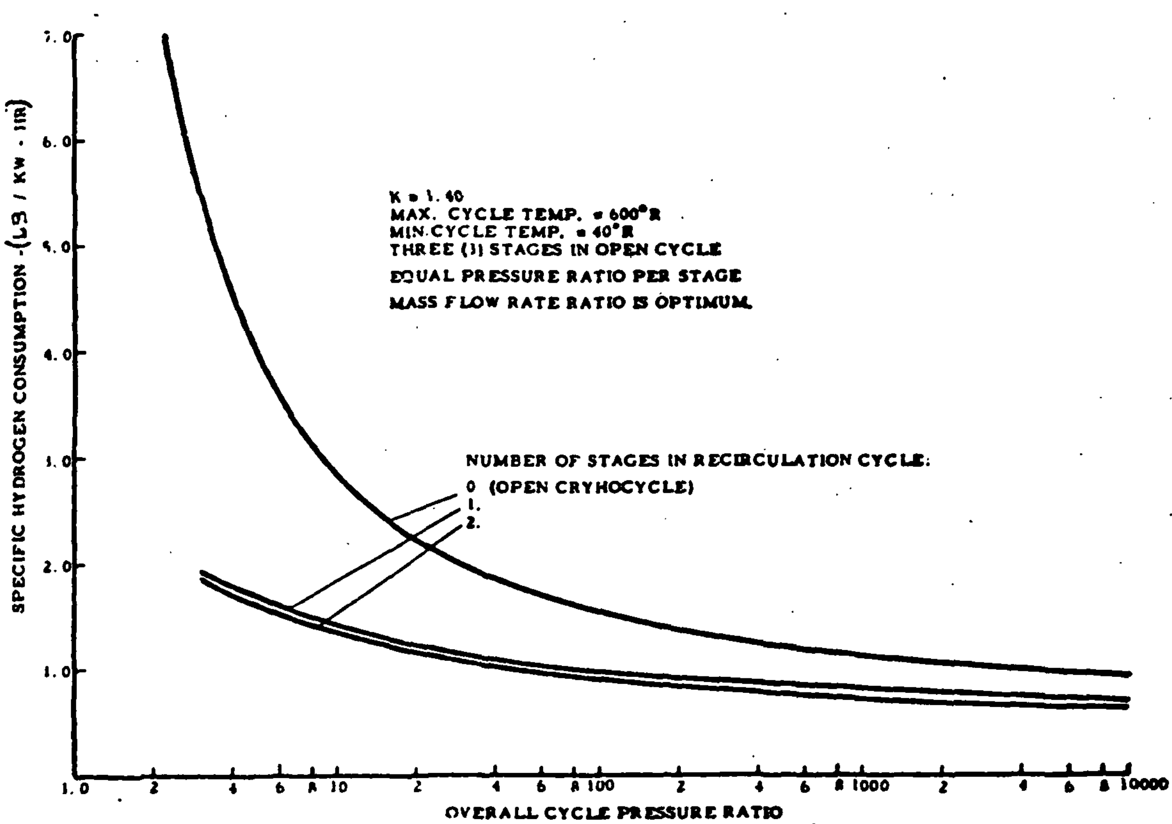


Fig. 3-24 Variation of Specific Hydrogen Consumption with Number of Recirculation Stages

half the open-cycle value, but to raise the minimum sustaining heat requirement and lower the maximum non-electrical cooling capacity.

3.3.4.5 Expander Type. Both reciprocating and turbine expanders may be used. The reciprocator is heavier, more efficient, and when properly designed has better performance away from the design point. Turbine expanders are best suited to shorter-duration higher-power missions where the total machinery weight will represent a relatively larger portion of the total machinery plus hydrogen weight.

3.3.4.6 Summary of System Parameter Choices

(1) Multistage expansion is beneficial. At least two stages should be used. There is a minimal benefit to be gained from more than four stages.

(2) SHC falls with increasing overall pressure ratio, but little further reduction is found above ratio of about 100 to 1.

(3) The SHC is approximately inversely proportional to the maximum cycle temperature.

(4) By using a recirculation loop, the SHC can be reduced by almost 50 percent. However, the maximum available external heat-load cooling capacity is greatly reduced and the minimum heat input to sustain the cycle is increased.

(5) Systems using turbine expanders will show lower machinery weight and higher total weight than systems using reciprocating expanders.

3.3.4.7 Parametric Data. The Cryhocycle textbook contains some data for machinery weight and SHC for three typical systems. These data assume realistic practical values for expansion efficiency, conversion efficiency, and pumping power. The three systems are as follows:

(1) Two-stage reciprocating expander
Recirculation loop
No pump (super critical storage)

(2) Three-stage reciprocating expander
Recirculation loop
High-pressure pump (subcritical storage)

(3) Four-stage turbine expander
No circulation loop
Pump (subcritical storage)

These systems by no means represent the totality of possible arrangements. Also, the performance data will vary from application to application. The data given are to be used solely for preliminary design purposes.

Figure 3-25 shows machinery weight as a function of electrical power output. These data include all hardware weight except that of the storage tank. This includes two power systems and a crossover module. The crossover module is a system component which makes possible the flow of hydrogen and coolant to an alternate power unit in the event of a failure of the primary unit. The curves show the classic comparison of rotary and reciprocating devices. At higher powers, the turbine has a clearly lower weight, while at low powers the reciprocator is superior. This effect is a result of the virtual impossibility of preventing a sharp fall in turbine efficiency as the size is reduced.

In recent studies performed by Grumman Aerospace Corp., Cryhocycle machines were re-evaluated. The results of those studies indicate that the machines are lighter than those indicated in Fig. 3-25. A typical weight breakdown is shown in Table 3-4. The weights in the table are for a machine similar in function to the three-stage reciprocator with recirculation shown in Fig. 3-25. If two machines with a crossover module is assumed, the total weight of the Grumman machine is 328 lb. This same machine as estimated by Sundstrand several years ago, as indicated by the curve is 560 lb. These two values probably establish reasonable upper and lower bounds.

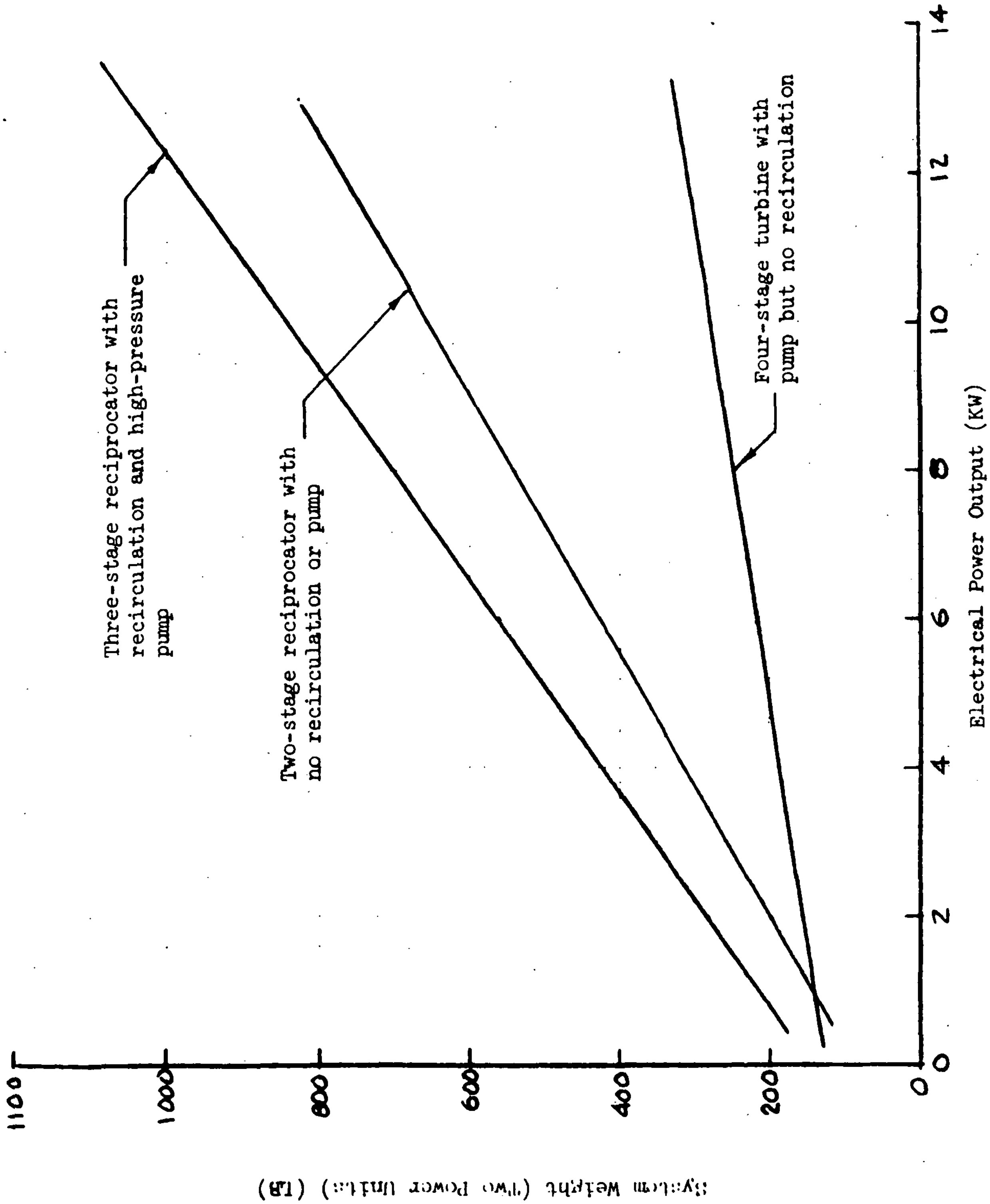


Fig. 3-25 Machinery Weight vs Electrical Power Output

Table 3-4

CRYHOCYCLE WEIGHT SUMMARY

(GRUMMAN CORP. ESTIMATES)

6KW OUTPUT PER MACHINE

Reciprocator, Including Compressor		65
Gearbox		10
Generator (6KW)		15
Hydrogen Pump with Motor		15
Combustor/Heat Exchanger		15
Heat Exchangers		29
Precooler	6.5	
Regenerator	4.0	
Recuperator	1.5	
Preheater	3.5	
Reheaters and Aftercoolers	13.5	
Controls		10
Total Per Machine		159
Crossover Module for Three Machines		10

Figures 3-26, 3-27 and 3-28 show SHC as a function of the external-to-electrical heat-load ratio; as explained earlier, there is an upper limit to the amount of external cooling that can be provided by the Cryhocycle, and this limit is dictated by the difference in the enthalpy of the hydrogen at points 10 and 11, Fig. 3-19 and 3-22. This upper limit is shown in the figures. The lower external cooling capacity of the systems using recirculation is apparent. Each of the SHC figures shows a band of possible SHC values, reflecting the generalized nature of these data. In the case of System 1, the band is broadened somewhat as a result of the use of a super-critical storage system. The supply pressure will fall as hydrogen is withdrawn from the tankage and expander efficiency will vary accordingly. The mission average SHC will therefore depend upon whether the electrical load is distributed uniformly over the mission, or whether large outputs are required at the mission beginning or end. In the case of System 3, a range of SHC values is shown, reflecting the rapid fall or turbine efficiency with decreasing capacity. Another effect shown on the SHC figures is that the upper limit of external-to-electrical heat ratio increases with increased SHC, since the external heat-load cooling capacity is directly proportional to working fluid-circulation rate.

As noted previously, there is also a minimum external heat input required to sustain the cycle.

This necessary sustaining heat input is proportional to the difference between the enthalpy of the hydrogen as it leaves the system, points 10 or 11, and the enthalpy at the storage condition, point 1. The actual heat required will be equal to the mass-flow rate times this enthalpy difference. Thus, the higher efficiency systems will have a relatively lower minimum sustaining-heat requirement per unit electric-power output because of the lower mass-flow rate required to produce the electric power. However, systems using recirculation have a higher minimum requirement, because the lowest venting temperature, point 10, is much higher than in the case of no re-circulation (compare Figs. 3-19 and 3-22). Figure 3-29 has been prepared to show the acceptable operating regimes more graphically.

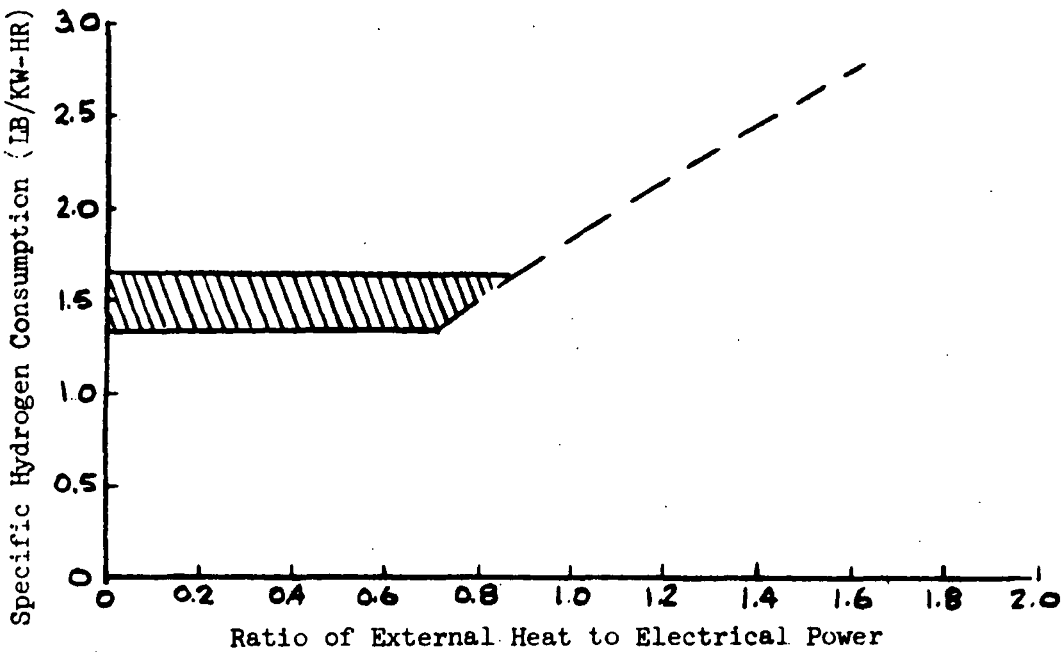


Figure 3-26 Specific Hydrogen Consumption for System 1

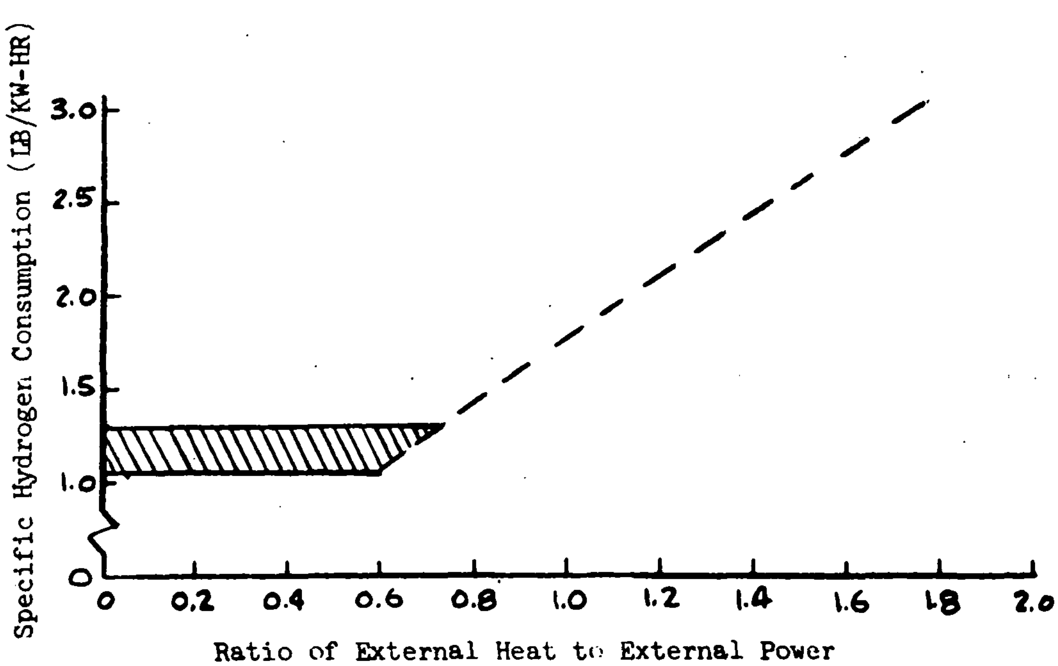


Figure 3-27 Specific Hydrogen Consumption for System 2

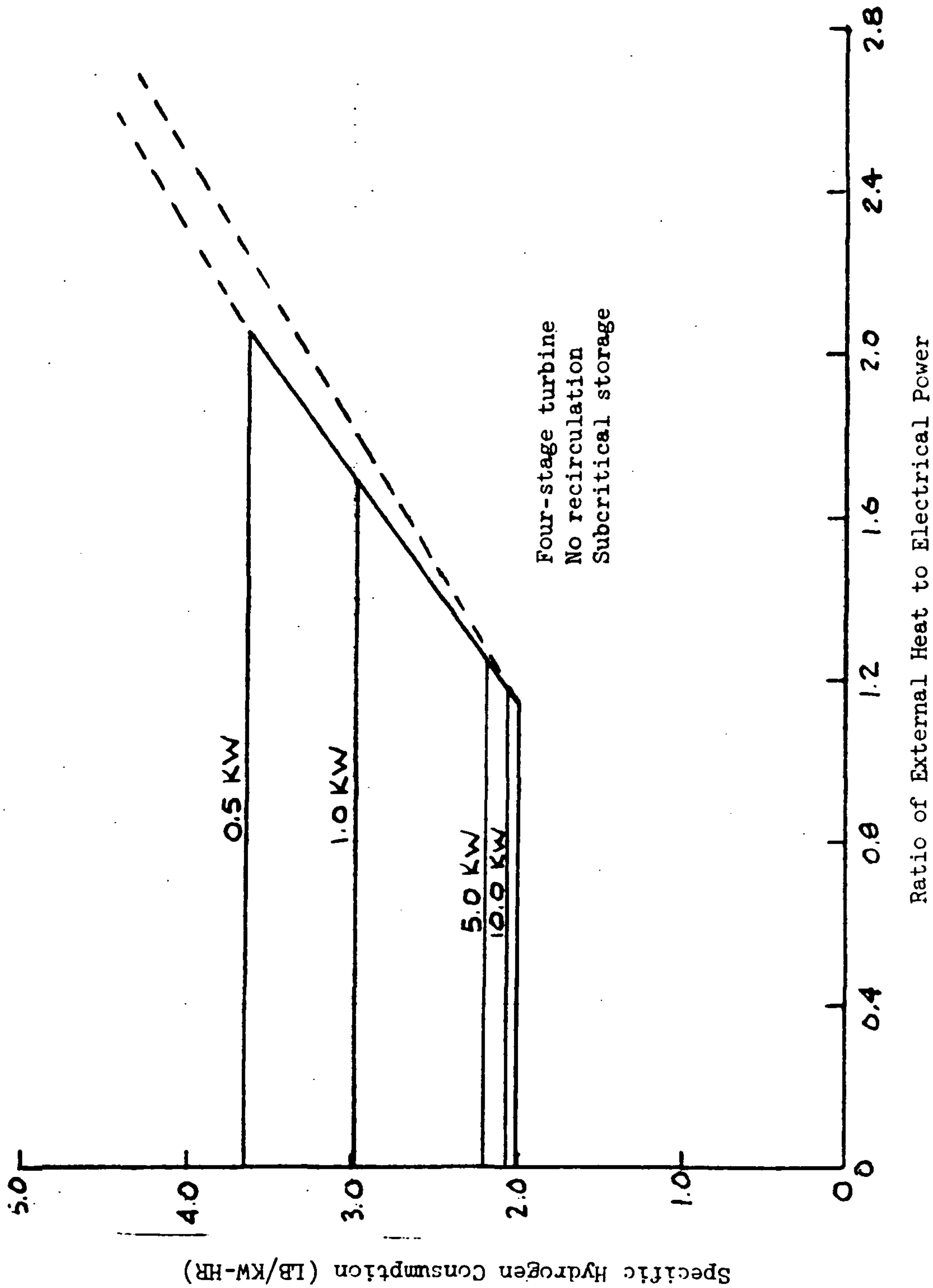


Fig. 3-28 Specific Hydrogen Consumption for System 3

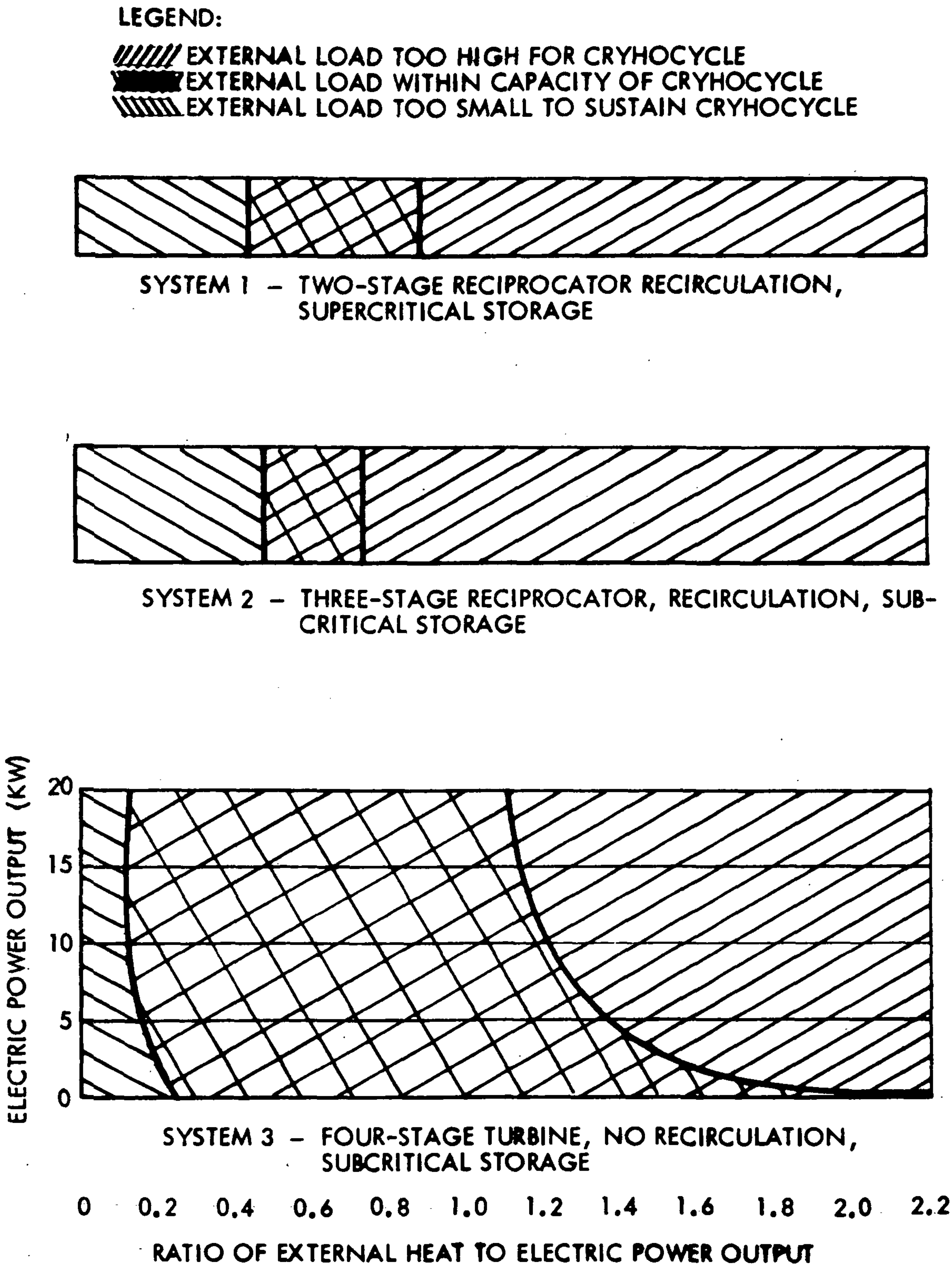


Fig. 3-29 Operating Regimes of Various Cryhocycle Systems

3.3.4.8 Cryhocycle Control Techniques. The Cryhocycle is fundamentally an electric generator system, and the basic parameter to be controlled is the speed. The speed will rise or fall in response to the variation in load torque, which in turn will depend on the magnitude of the electrical load. Several control techniques may be considered; all include a speed variation sensor whose output can be used to vary the following parameters:

(1) Working fluid flow rate can be varied by:

   (a) a variable bypass valve to direct a portion of the inlet flow rate directly to the exhaust,

   (b) a variable inlet flow throttling valve, or

   (c) variable inlet geometry for the expander.

(2) Working fluid inlet pressure can be varied by varying the speed or displacement of the liquid supply pump.

(3) The heat input to the expander can be varied by bypassing the electrical load coolant loop flow through the pre-heater and reheaters.

The usual technique used for control would be (1b) for a turbo expander and (1c) for a reciprocator. The time constants usually associated with speed control by these techniques are substantially less than one second. However, variation in operating conditions will also result in temperature transients in the main heat exchanger, external load heat exchanger, and electrical load coolant loop. The external load exchanger is largely independent of the main Cryhocycle system, and its temperature transients can be minimized by sensing the gas temperature and pressure at the system exhaust points and controlling the proportioning valve accordingly. The other two exchangers, however, will pass through a period when they will be temporarily unbalanced. Because of the relatively long time required for coolant or working fluid to complete one pass around their respective circuit, the time constant for the transients in these exchangers will be of the order of several seconds. Although this is a significant control system design problem, it can be

readily solved. The primary effect of the long temperature transient time constant will be to produce temperature excursions at the expander and electrical load. If these excursions are not between acceptable limits, they can be damped by adding thermal cpacity. If this is not adequate, a more sophisticated system of heat management must be employed. For example, part of the cold vent gas could be directed to minimize temperature rise at the load or expander. In any event, the overall design of the control system will not present any insurmountable or unusual problems. It is simply a case of determining the time constants of a particular application, the magnitude of expected output variations, the permissible ranges of operating conditions, and then selecting suitable hardware to perform the necessary control tasks.

3.3.4.9 Cryhocycle Off-Design Performance. The cryhocycle will have two major functions to perform - generating electricity and cooling external heat loads. In general, the magnitude of these tasks will not vary in unison and thus two separate control systems will be necessary. Since the cooling capacity and power output are directly related, however, some form of logical interrelation of these controls will be possible. Design of these controls should be straightforward and their details are not relevant at the preliminary design stage.

As noted above, the output of the expanders can be regulated in several ways. The most elementary method is to vary the inlet mass flow by means of a throttling valve. This will reduce or increase the output around the design point with some loss of efficiency. A second method is to vary the expander geometry in such a manner that the working fluid consumption is varied. In the case of the reciprocator, this can be effected by varying the inlet valve cutoff point. In the case of the turbine, the admission area is varied so as to maintain design velocity levels. Again, this will result in some loss of efficiency. However, if throttling and geometry variation are used concurrently, the loss of efficiency at off-design conditions can be greatly reduced.

It is not possible to show generalized plots of SHC versus percent load without pre-supposing a particular control philosophy. However, the Cryhocycle textbook shows such a plot for a particular type of control technique. The plot is reproduced in Fig. 3-30. The system is a three-stage reciprocator with one stage of recirculation. To obtain power outputs below the design value, the inlet gas pressure to the expander is reduced by means of a throttle valve. To obtain power outputs above the design level, unthrottled high pressure gas is admitted to the first stage and is then vented. Throttled high pressure gas is admitted directly to the second stage, which operates in series with the third stage as in normal operation. These control techniques are simple, but relatively inefficient.

Regulation of the gas flow rate in response to changes in electrical power demand will vary the cold gas flow rate at the exhaust point. If the external load remains reasonably constant during these electric power changes, the position of the flow proportioning valve will have to be adjusted to maintain a constant mass flow rate through the external load heat exchanger. If the electrical load is reduced, for example, a greater percentage of the venting gas will be directed through the external load heat exchanger. If the electrical power is reduced past the point where all of the vent flow is passed through the external load exchanger, additional cooling in the form of direct liquid hydrogen boil-off must be used. On the other hand, if the electrical power demand increases, the working fluid mass flow rate will be increased and a lesser percentage of the total flow will be passed through the external load exchanger.

Other comparisons of specific hydrogen consumption are shown in Fig. 3-31. These estimates were made by Grumman Aerospace Corp. and apply to the machine for which the weights are shown in Table 3-4. The minimum specific hydrogen consumption is shown to be about 1.3 lb/KWhr for the Gumman machine, whereas it was indicated to be about 1.12 in the Sundstrand Textbook. It is estimated that the SHC of 1.3 is a more realistic value.

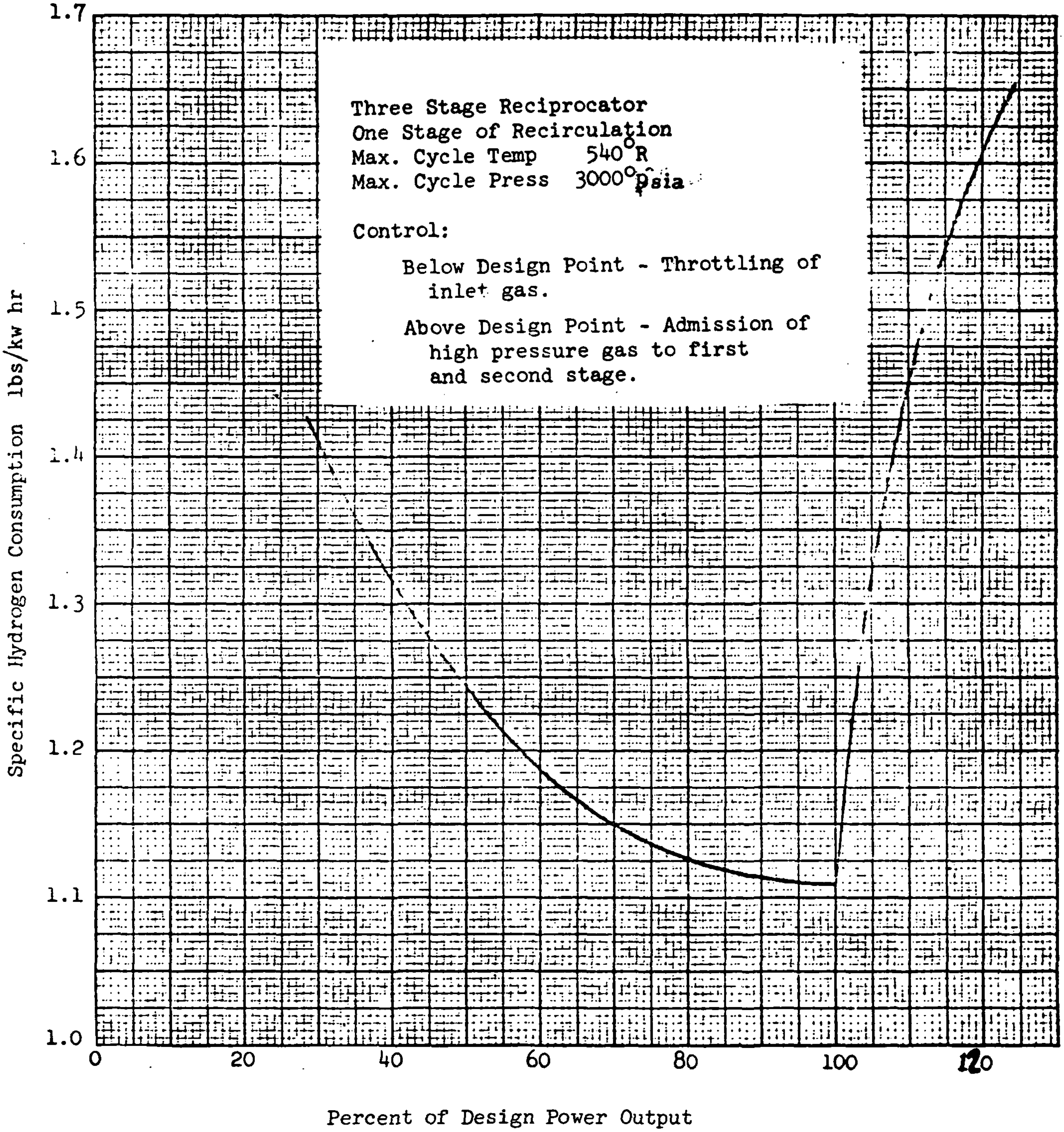


Fig. 3-30 Hydrogen Consumption at Off Rated Power

3.3.4.10 Summary. The foregoing material was presented to provide an understanding of what the Cryhocycle is and what some of its characteristics and parameters are. The Cryhocycle does provide a means of supplying power to the vehicle and simultaneously rejecting electrical and metabolic heat loads.

During the earlier periods of the Space Shuttle definition phase, when large quantities of hydrogen were being stored on the vehicle for propulsion purposes, the Cryhocycles seemed to have a natural application. With the removal of most of the hydrogen, however, the case for using a Cryhocycle weakened somewhat. However, there seemed to be some potential uses for the machine in the currently configured Space Shuttle, and a set of comparisons studies for the machine were performed. These studies are discussed in Section 3.3.5.

For the power requirements and the duration being considered for the Space Shuttle, the best type of Cryhocycle system appears to be a three-stage reciprocator with hydrogen recirculation. While the machine is heavier than a turbine-type of machine, the specific hydrogen consumption is less and therefore the overall system is lighter than a turbine expander system. On the other hand, the operating range, in terms of external heat to generated power required for operation, is smaller for the reciprocator with recirculation. If growth to longer missions is also considered, the reciprocator machine has an advantage. For the present, it appears that a three-stage reciprocator expander with hydrogen recirculation would be favorable.

3.3.5 Comparison of Cryhocycle and Baseline System for Orbital Operation

A weight comparison of a baseline system and a Cryhocycle system was made. Each system provides all of the power generating and cooling required by the mission. The baseline system consists of:

- Fuel Cells
- Radiators
- Freon Cooling Loop
- Hydrazine APU
- Dedicated Hydrogen Cooling System

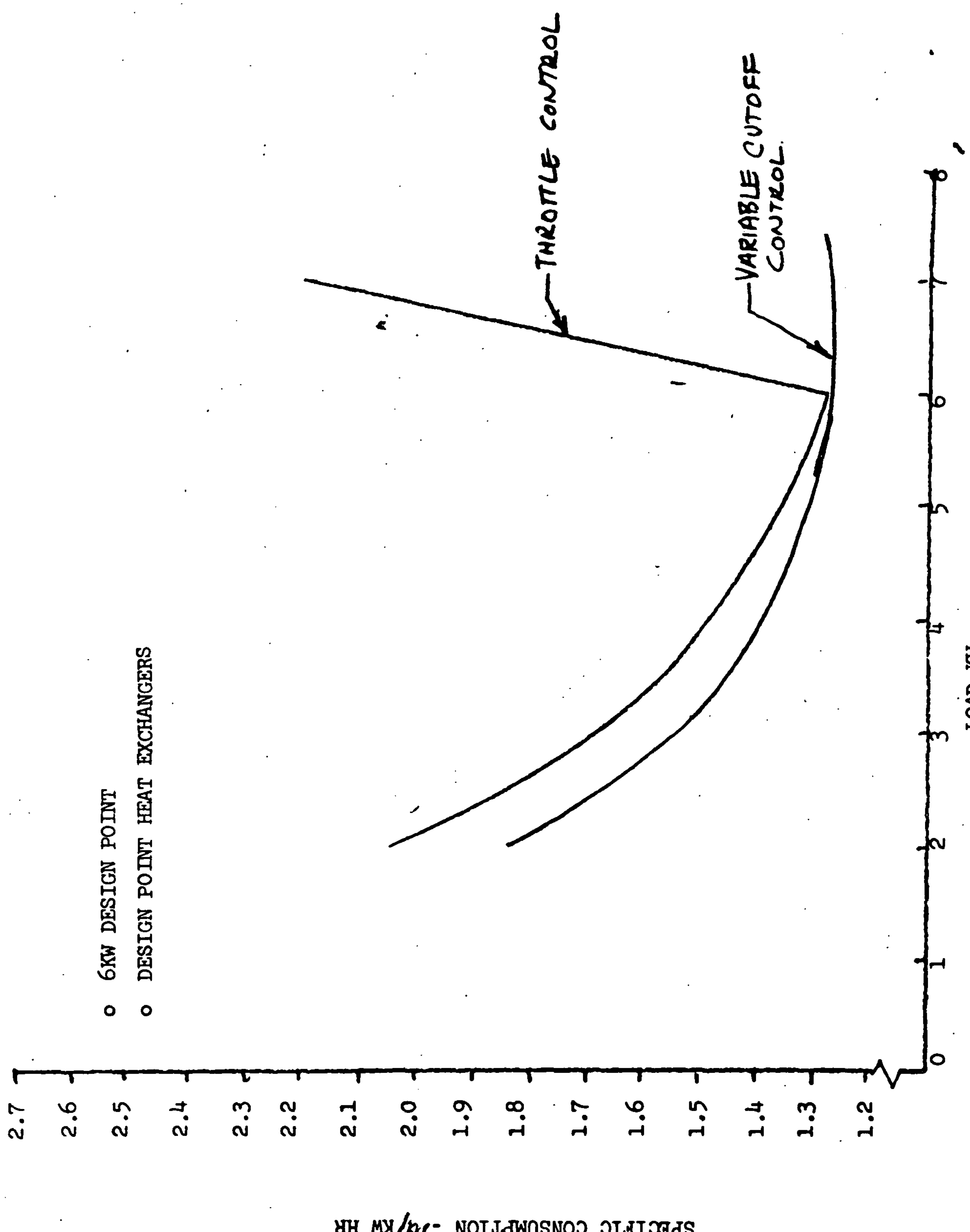


Fig. 3-31 Specific Hydrogen Consumption vs Load

The Cryhocycle system consits of:

- Cryhocycle
- Freon Cooling Loop
- Hydrazine APU
- Supplemental Hydrogen for Additional Cooling

Consideration was also given to including in the comparison a system that utilizes a Cryhocycle for the APU function. However, because of the large and transitory nature of the hydraulic power requirements, it was concluded that a Cryhocycle of significantly larger size would be required than that employed for orbital operations, and that a oxygen-hydrogen gas generator would probably have to be added. A system like this begins to resemble an oxygen-hydrogen APU more than a Cryhocycle, and therefore no analyses were conducted.

The Grumman Aerospace Corp. conducted a study on the Cryhocycle System for the NASA/MSC and investigated the system in considerable more detail than is possible here. During the early phases of that study a power profile was agreed upon and, in order to provide some consistency, that same power profile is used in this comparison. The reference profile is shown in Fig. 3-32. This is not the same profile that was used to establish the heat loads shown earlier in this report; however, on an average basis, the difference is small and insignificant differences arise in the comparison.

This profile indicates a minimum power of 5.2 KW and a maximum of 10.37 KW. The total energy required by the fuel cell or its alternate is 750 KWhr, including 5 minutes of prelaunch operation.

The energy level was used as the references for these comparison studies; however, subsequent studies on the power requirements of specific avionics components indicate that an energy level as much as three times this value may be required. The influence that this higher power requirement has on the comparison will be discussed in the last part of this section.

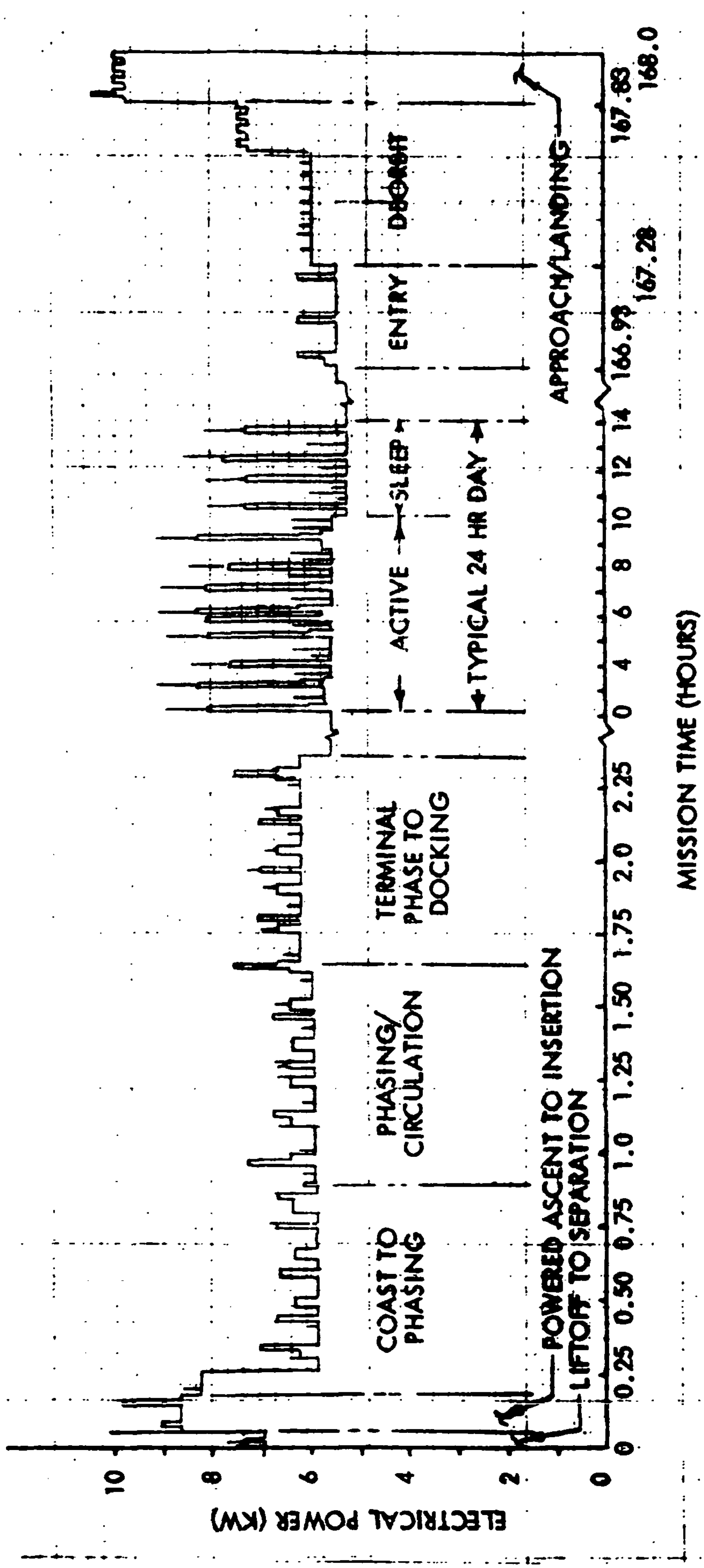


Fig. 3-32 Orbiter Electrical Power Profile

The power requirements for hydraulic systems and additional alternators during ascent and reentry are given in Table 3-5.

3.3.5.1 Baseline Systems. The baseline system consists of:

- Fuel cells and cryogenic storage systems
- Space Radiators
- Freon Cooling Loops
- Hydrazine APU
- Dedicated hydrogen reentry cooling system.

The basic groundrules and assumptions are as indicated here.

- Storage tanks will be the subcritical type, with the fluid stored at as low a pressure as is compatible with the using system requirements.

Fuel Cell	-	120 psia	-	no pump
APU	-	30 psia	-	pump required
Cryhocycle	-	30 psia	-	pump required

- Supercritical storage tanks for the fuel reactant will also be considered.
- Zero-g acquisition systems will be available and are employed where required.
- Where pressurization is required, helium will be used as the pressurant except for fuel cell supply tanks which will be self-pressurized from heat feedback.
- Three fuel cells shall be used; one shall operate and two are on standby.
- Three APUs shall be used, each with full power capability but only two operating at 1/2 power and the third on standby.

Table 3-5

ASCENT AND REENTRY POWER REQUIREMENTS

Period	Power (hp)	Turbine Discharge Pressure (psia)	Time (Min.)
Checkout	32	15	12
Boost	32	10	3.5
Coast	32	5	0.2
Insertion	32	5	3.25
Reentry	32	5	50
Reentry	75	5	25
Cruise	147	10	1.25
Cruise	32	10	1.25
Cruise	85	15	1.33
Approach	85	15	.675
Flare	105	15	.225
Touchdown	135	15	.45
Touchdown	32	15	.45
Go-around	150	15	1.25
Go-around	135	15	2.50
Go-around	105	15	1.25

3.3.5.1.1 Fuel Cells. The fuel cell system, which supplies primary power throughout the vehicle flight profile, is assumed to be the type that has the characteristics of the Pratt & Whitney Aircraft Corp. fuel cells. The assumed characteristics are shown in Table 3-6. It is assumed that all internal cooling and plumbing sufficient for the operation of the fuel cell is included in the specific weight of 35 lb/kW. The additional plumbing and cooling shown in the table is for fuel cell to tank supply and cross-over and for coolant loop to module heat removal. The power output of 6.5 kW was based on the power profile shown in Fig. 3-32.

A relatively low supply pressure was selected to minimize the storage system weight. The cryogens are supplied at a nominal pressure of 120 psia, which will probably preclude the use of jet pumps for coolant and water separator flow on the fuel cell modules. However, electric driven pumps can be used.

3.3.5.1.2 Storage System. The total reactant required is based on a specific reactant consumption of 0.86 lb/kWhr for 750 kWhr. With a 20-percent reserve the total is 722 lb of reactant, or 80 lb of hydrogen and 642 lb of oxygen. The storage characteristics are shown in Table 3-7. A subcritical storage system was selected to minimize the weight. The pressure of 120 psia was selected to ensure that a nominal pressure of 60 psia could always be regulated to the cell stack. Some weight savings could be realized if the pressure were reduced, since the hydrogen and oxygen tank wall thicknesses are about 0.050 and 0.037-in. respectively. However, this is a relatively small gain of 3 lb for 10 psia for both tanks.

As an alternate approach for storing the reactants, a superciritcal tankage system was considered. For the loaded hydrogen weight of 95 lb, the tank system weight is about 140 lb. For a loaded oxygen weight of 661 lb, the tank system weight is about 155 lb.

As indicated in Fig. 3-33, the supply system withdraws fluid at whatever state exists at the exit port and passes it through the vapor shield to a cryogen-Freon heat exchanger. The heated fluid is diverted back past the

Table 3-6

FUEL CELL CHARACTERISTICS

Weight lb/kW	35
Power - Peak kW	10
Power - Steady kW	6.5
Voltage Regulation 0 to 6.5 kW	±6%
SRC lb/kW hr	0.86
Operating life hr	10,000
Reactant Supply Pressure psi	100
Reactant Purity	Propulsion grade
Heat rejection	Coolant loop
Heat rejection wt lb	74
Plumbing lb	147

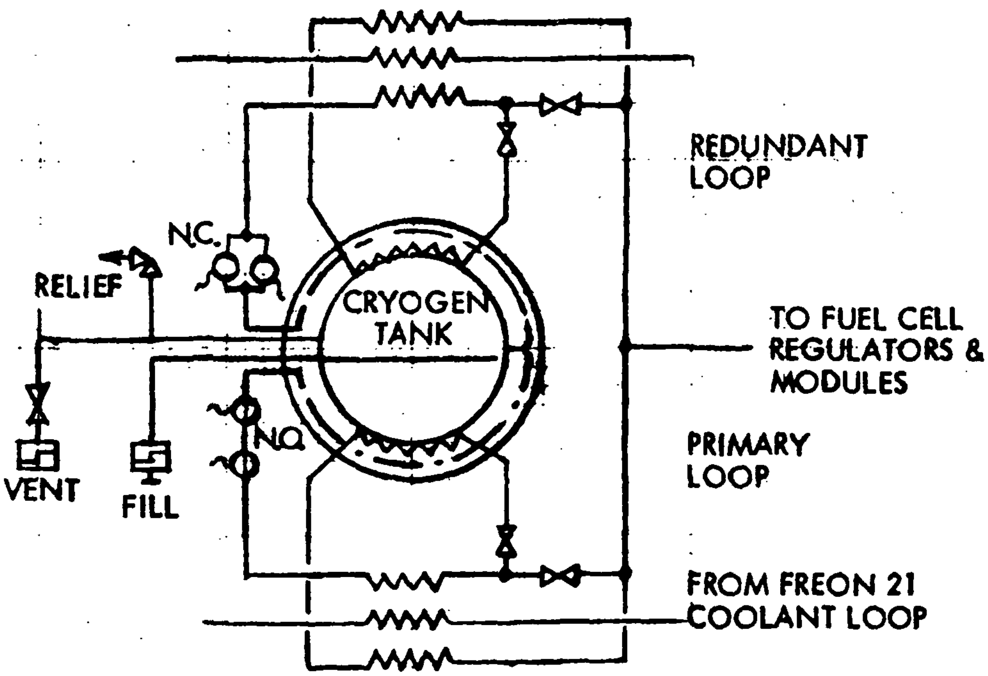


Fig. 3-33 Typical Subcritical Cryogen Fuel Cell Supply System

Table 3-7

REACTANT STORAGE TANK CHARACTERISTICS

		H_2	O_2
Tank			
Nominal pressure	psia	120	120
Control band	psia	± 10	± 10
Relief valve setting	psia	± 10	± 10
Sphere size	in	42	32
Material		2219 al.	2219 al.
Maximum Mat'l temp at operating pressure		540°R	540°R
Safety factor		2	2
Weight	lb	40	20
Insulation			
Double Aluminized Mylar			
Thickness	in	1.5	1.5
Weight	lb	19	12
Vacuum Jacket			
Honeycomb min material thickness		0.01	0.01
Weight	lb	29	18
Valves, heat exchangers, lines			
Weight	lb	53	50
Cryogens			
Usable weight (incl. 20% reserve)	lb	80	642
Residual weight	lb	15	19

tank if the pressure is low, or directly to the fuel cell modules if the pressure is high. If the flow is passed over the tank it is reheated in the heat exchanger on the way to the fuel cell module. The primary loop is normally operating; however, in case of a failure of the primary loop or the associated Freon cooling loop, the redundant loop is activated.

The tanks are vacuum jacketed and insulated, so that no boiloff is experienced throughout a normal mission.

3.3.5.1.3 Radiators. A thermal analysis was performed for the alternate concepts vehicle studies to establish radiator performance and size required for a range of design environmental conditions applicable to the orbiter. Since the Space Shuttle may experience a wide range of mission conditions, a maximum heating orbit environment was considered. The following assumptions were made:

- The radiator is located on the orbiter so that its view to space is unobstructed; i.e., the radiator panel has a view factor to space of 1.0. This could be a panel on the top of the orbiter or on the inside of a fully opened door. The radiator panel is assumed to radiate from one side only.
- Orbit altitude is 270 nm.
- External environment heat rates are for a high β angle orbit, giving a combination of maximum time in sun and albedo. The angle β is defined as the acute angle between the earth-sun line and orbit plane.
- With low α_s/ϵ surfaces used on the radiator, the maximum external heating condition is when the radiator faces the earth.
- The radiator has a low α_s/ϵ surface (OSR). Values of α_s considered in the study were 0.055 and 0.09, resulting in α_s/ϵ ratios of 0.07 to 0.11. The lower value is achievable

with an ideal, undegraded OSR surface on flat panels. The higher value is indicative of actual installations of small mirrors (1-1/2 by 1-1/2 in.), where edge effects and some degradation is inherent. Values of about 0.1 to 0.11 have been demonstrated on previous flight programs.

External environment heating rates, which include solar, albedo, and earthshine, actually vary with orbit position, but were averaged over the orbit period. This is common practice in sizing radiators. The averaged heating rates were:

- Direct solar and reflected solar (albedo) = $65.5 \frac{\text{Btu}}{\text{hr-ft}^2}$
- Earthshine = 60.0 Btu/hr-ft^2

The following parameters were selected from the analysis and are representative of anticipated values for the orbiter:

- Coolant - Freon 21
- Coolant specific heat = f (Temperature)
  (Cp = 0.22 at =240°F and 0.28 at 360°F)
- Flow rate = 2130 lb/hr
- Solar absorptance, α_s = 0.09
- Infrared emittance, ϵ = 0.8
- Radiator area = 800 ft^2
- Heat rejection rate; 40,000 Btu/hr
- Fin efficiency = 1
- T_{out} = 35°F
- T_{in} = 110°F

The radiator configuration was assumed to consist of several panels on the outside of the payload bay doors. Each panel has the capability of being operated individually and is about 20 ft^2 on each half, so that when a panel is open there is 40 ft^2 of radiator area exposed. There are 10 panels on each of the two payload doors, so that a total area of 800 ft^2 is achieved. The weight of these panels are assumed to be 1.5 lb/ft^2, including the

actuator, fluid, and supporting structure. The total weight of the radiator is 1200 lb.

Both the area and unit weight are considered to be conservatively large and one could expect to be able to reject considerably more heat for a radiator weight of 1200 lb.

3.3.5.1.4 Freon Cooling Loop. The Freon cooling loop is part of the environmental thermal control system. It provides the heat transport mechanization to remove heat from the cabin water loop, from the fuel cell, and from equipment located outside the cabin. Estimates of the system weight are shown in Table 3-8.

3.3.5.1.5 APU System. The APU is supplied with hydrazine via a high pressure pump that draws from a low pressure tank. The tank is pressurized to 30 psia with helium from a high pressure storage system. The hydrazine is withdrawn from the tank via a surface tension acquisition device which needs to function only during the orbital start-up period and during the early phases of reentry, Weight estimates are shown in Table 3-9.

3.3.5.1.6 Cooling System. During the ascent the cooling loads are estimated to be:

Prelaunch	33,121	66,189
Ascent to injection	33,068	

Of this amount approximately 48,960 Btu are generated at the APU. If the APU is allowed to heat from 70°F to 170°F, approximately 42,500 Btu can be absorbed. If it is assumed that no APU cooling will be provided during ascent, then about 17,229 Btu are generated from other sources.

If the Freon and water systems are permitted to heat 20°F, then about 11,000 Btu can be absorbed. This leaves approximately 7,000 Btu to reject, which can be done easily with water at the higher altitudes or by using ascent tank cryogens. Therefore, no penalty will be assigned for cooling during ground hold and ascent.

Table 3-8

FREON SYSTEM LOOP WEIGHT*

Water Sublimator	(1)	20
GSE heat exchange	(1)	25
Cryo hydrogen heat exch	(2)	10
Intercooler	(2)	80
Accumulators	(4)	20
Pump	(4)	25
Cold plates/heat exchangers		250
Valves/Reg/Controllers		150
Plumbing & Fittings		75
		655
Fluid		400
		1055 lb

*Based primarily on alternate concepts study weight estimates

Table 3-9

APU SYSTEM WEIGHTS

APU (3)	366
Alternators	10
Sump, oil, etc.	207
Oil cooling	10
N_2H_4 Tank	21
He pressurization system	12
N_2H_4 pumps	6
Plumbing and Valves	15
Acquisition System	5
	652
Hydrazine (110 lb usable)	1206
	1858 lb

During orbital operations the cooling is accomplished by the radiator, with temporary peak cooling loads being handled by water sublimation.

The heat loads estimated in Section 2.3.1 were used for determining the amount of cooling required for the reentry portion of flight. A total of 363,000 Btu was assumed to be rejected. This was based on the assumption that heat could be rejected to the air during the last 12 minutes of flight and that the ECS and APU systems could heat up to 20°F and 100°F, respectively. The total hydrogen required for cooling would be 191 lbs. A reserve of 10 percent was added to this and with the residuals the total loaded is 220 lb. The hydrogen is stored subcritically in a 55-inch spherical tank with a minimum wall thickness of 0.025 in. This condition, with a safety factor of 2, results in an allowable pressure of 61 psia. The maximum pressure is estimated to be 50 psia if the tanks are locked up 2 minutes before liftoff. No venting is required and since no pump is used the fluid quality is not important and therefore no acquisition or pressurization system is required. The summary of the cooling system weight is shown in Table 3-10.

3.3.5.1.7 Summary of Baseline System. The total weight for the baseline system is shown in Table 3-11. This system provides all of the primary power and cooling required throughout the mission profile.

3.3.5.2 Description and Sizing of the Cryhocycle System. The Cryhocycle System that is compared to the baseline system consists of the following subsystems:

- Cryhocycle
- Freon Cooling Loop
- Hydrazene APU
- Supplemental Hydrogen for additional cooling during reentry

The basic groundrules and assumptions are as follows:

- Size hydrogen supply for cooling orbit only for maximum heating orbit.

Table 3-10

SUMMARY OF COOLING SYSTEM WEIGHT

H_2		220
Usable	191	
Reserve	20	
Residual	9	
Tank		37
Ins (incl purge bag)		34
Plumbing		30
Purge		8
Dry Weight		109
Wet Weight		325 lb

Table 3-11

WEIGHT SUMMARY OF BASELINE SYSTEM

Fuel Cell System		1146
Fuel Cells	684	
Plumbing	147	
Heat rejection	74	
H_2 Tankage	88	
O_2 Tankage	50	
Plumbing	103	
Radiator System (including fluid)		1200
Freon Coolant Loop		655
APU System		652
Cooling System		109
Fluids		
FC H_2		96
FC O_2		661
Freon		400
Cooling H_2		220
N_2H_4		1206
Total		6345 lb

- Size oxygen supply for supplying additional heat during coldest orbit.
- Cryhocycle machine is a 3-stage reciprocator, with one stage of recirculation operating at 3000 psi and 540°R. Oxygen is reacted in a catalytic heat exchanger. Three expanders are used for redundancy purposes.
- Peak power is supplied by APU during ascent and reentry and by peaking batteries during orbital operations.

Additional assumptions will be listed as they are required for the development of the various components and subsystems. Each subsystem will be discussed in the order given above and the weights developed for each. At the end the weights are summarized and compared with the baseline system.

3.3.5.2.1 Cryhocycle Machine. The description and function of the Cryhocycle has been discussed in detail by Sundstrand Aviation in their "Cryhocycle Text Book" and has been summarized in Section 3.3.4.

For the purposes of conducting a comparative analysis between the two types of systems (the baseline system versus the Cryhocycle system) it is necessary to select one type of machine and feed system. After considering the previously-discussed points and performing a preliminary evaluation of the trade-off between the machine weight and the SHC, a three-stage reciprocator with one-stage of recirculation was selected. This machine is heavier than a turbine expander machine but this is more than made up by the lower SHC.

To obtain the machinery weight, the upper parametric curve shown in Fig. 3-25 was used. The SHC for this machine depends somewhat upon the method of control.

As noted in Section 3.3.4, the output of the expanders can be regulated in several ways. The most elemetary method is to vary the inlet mass flow by

means of a throttling valve. This will reduce or increase the output around the design point with some loss of efficiency. A second method is to vary the expander geometry in such a manner that the working fluid consumption is varied. In the case of the reciprocator, this can be effected by varying the inlet valve cutoff point. For a three-stage reciprocator with one stage of recirculation, the inlet gas pressure to the expander is reduced by means of a throttle valve to obtain power outputs below the design value. To obtain power outputs above the design level, unthrottled high-pressure gas is admitted to the first stage and is then vented. Throttled high-pressure gas is admitted directly to the second stage, which operates in series with the third stage as in normal operation. These control techniques are simple, but relatively inefficient.

To establish the size of the Cryhocycle machine, the SHC, and peaking batteries, several preliminary iterations were made in these three parameters. A nominal power of 6.7 kW was selected. This is based on the assumption that a 10 percent overpower is available. The total weight of the machine, including all heat exchangers, controls, and crossover control for supplying two expanders is 620 lb. For three expanders, heat exchangers, and crossover, the system weight is estimated to be about 910 lb. The SHC for this machine is shown in Fig. 3-30.

The hydrogen consumption for each phase of flight and at difference power settings is summarized in Table 3-12; the total is 1149.3 lb. The power profile shown in Fig. 3-32 was used as a basis. A relatively small fraction of the time is spent in an overpower or significant underpower condition and the machine operates at a low SHC for most of the duty cycle. A few peak power points cause a total of 4235 W-hr to be expended by discharge and recharge of batteries. The batteries were sized for a 330 W-hr discharge and a 50 percent discharge depth. A 75 percent recharge efficiency was employed. For weight purposes, a type-29 rechargeable battery of 660 W-hr was used, with a weight of 110 pounds. Two batteries were assumed.

Table 3-12

HYDROGEN CONSUMPTION FOR CRYHOCYCLE POWER

Phase	Time	1 t (min)	1 Power (kw)	1 %	2 t (min)	2 Power (kw)	2 %	3 t (min)	3 Power (kw)	3 %	SHC 1	SHC 2	SHC 3	Total
Liftoff to Separation	0:02:27	2.5	6.7	100							1.11	-	-	.31
Powered Ascent to Insertion	0:07:32	7.5	6.7	100							1.11	-	-	.93
Coast to Phasing	0:43:32	22	5.8	87	15	6.7	100	1.5	7.3	110	1.12	1.11	1.6	4.6
Phasing/ Circulation	0:45:12	30	5.9	88	10	6.3	94	15	6.7	100	1.12	1.11	1.11	7.58
Term. Phase to Docking	0:42:23	30	6.1	91	10	6.7	100	2	7.1	105	1.11	1.11	1.45	5.0
Orbital Active	116:35:20	96 hr	5.6	83.5	11 hr	6.2	93	9.5 hr	7.3	110	1.12	1.11	1.6	794
		Battery recharge = 3710 w-hr at 5.6 or 83.5%												4.2
Sleep	48:00:00	36 hr	5.2	77.5	5 hr	6	90	7 hr	7.3	110	1.13	1.11	1.6	325
		Battery recharge = 525 w-hr at 5.2 or 77.5%												.6
Deorbit	0:21:0	17	5.4	80	4	6.1	91				1.13	1.11		2.2
Entry	0:33:0	23	5.8	87	10	6.7	100				1.12	1.11		3.7
Approach & Landing	0:10:0	10	6.7	100							1.11			1.2
Total consumption through expander														1149.3 lb

To show the influence on complete systems, the different Cryhocycle machine weight and SHC defined by Grumman Aerospace Corp. were also used. The data shown in Table 3-4 were used as the basis to estimate the weight of a Cryhocycle unit with three expanders, the associated heat exchangers, and a crossover modules. For a unit with an output of 6.7 kW, the weight estimate is 528 lb. The SHC associated with this machine is shown in Fig. 3-31 for both throttle-type control and variable cutoff-type control. The minimum SHC is about 16 percent larger than that shown in Fig. 3-30 and probably represents a more realistic estimate.

The total hydrogen consumption for the SHC is approximately 1344 lb for the throttled control case.

3.3.5.2.2 <u>Oxygen Requirements</u>. At an average power of 6.7 kW, the Cryhocycle has more than enough cooling capability to handle all of the orbital non-electrical heat load for a "hot" orbit ($\beta = 65°$); therefore, no additional heating must be added. It was estimated that the average net heat flux out of the cabin could be as much as 5000 Btu/hr, including the metabolic heat. The Cryhocycle requires 9250 Btu/hr non-electrical heat at 6.7 Kw to sustain operation. Therefore, the heat that must be added for a worst case cold orbit is 14,250 Btu/hr. Since it might be possible or desirable to operate the shuttle in a cold orbit altitude for the duration of flight (and more important, one would not want to limit the shuttle orbital operation on the basis of power), it was decided to size the heat addition requirements for a continuous 14,250 Btu/hr.

A different type Cryhocycle machine, such as a four-stage turbine with no recirculation, would permit a lower heat addition requirement. However, oxygen would still have to be used for some phase of the mission, so the added complexity of a oxygen hydrogen reactor would always have to be accepted. Once this penalty is accepted, it is preferable to increase the amount of oxygen used to reduce the SHC. This was done when selecting the more efficient Cryhocycle machine.

The heat addition was assumed to be achieved by a catalytic combustor-heat exchanger. The overall reaction was assumed to occur at near stoichiometric conditions. Since the points near the oxygen injection will be oxygen rich, heat is transported to the coolant to keep the overall temperature to a few hundred degrees hotter than the coolant. The reaction is a non-adiabatic one. The heat available was estimated to be 5100 Btu/lb of reactant at a mixture ration of 8:1. At these conditions the total additional oxygen and hydrogen required for a 164-hour period is 409 lb and 51 lb, respectively.

The hydrogen is stored along with the main Cryhocycle hydrogen supply. Separate additional tanks must be provided for the oxygen. Since a more or less worst case condition was postulated for determining the amount of reactants no reserve is added for this particular function. The total cryogens are shown in Table 3-13 along with the storage tank characteristics.

3.3.5.2.3 Cryogens Storage and Supply. The storage system consists of two cylindrical tanks for hydrogen and one spherical tank for oxygen. The characteristics for the cryogen load are shown in Table 3-13. For the larger cryogen load the weights are summarized in Table 3-14. Cylindrical tanks had to be employed for the hydrogen because of installation problems associated with a larger spherical tank (8.7 ft diameter for one tank). Each vehicle configuration would have its own unique installation arrangement. The installation restrictions were based on the Lockheed 040 baseline orbiter. A penalty in tank weight results from utilizing cylindrical tanks but it is a reasonable and expected penalty incurred by the use of large amounts of hydrogen.

The hydrogen is stored subcritically and is supplied to the Cryhocycle pump via a surface tension acquisition device. Helium is used for pressurization during orbital operation. To save weight, vacuum jackets were not assumed. The tanks are insulated with multilayer insulation and purged with helium during atmospheric flight and evacuated to orbital vacuum during orbital flight. The high heat rates that occur during the atmospheric operation are accepted as pressure rise. During ground hold and ascent phases the hydrogen

Table 3-13

CRYOGENS STORAGE CHARACTERISTICS

Tanks	H_2	O_2
Number	2	1
Shape	Cylinder with hemi-spherical ends	Spherical
Diameter (in.)	48	27
Length (ft.)	15.2	--
Operating pressure (psia)	35	150 (max.)
min-gauge	0.025	NA
Safety factor	2	2
Weight (each) (lb	142	21
Insulation thickness (in.)	1.5	0.5
Insulation weight (lb)	115	3
Acquisition System	Tube Channels With Surface Tension Heads	--
Acquisition System Weight (lb)	22	--
Inner Tank Dia. (in.)	18	--
Inner Tank Weight (lb)	5	--
Valves & plumbing Weight (lb)	74	27
Cryogen		
Cryhocycle Expansion (lb)	1149	
Heat Generation (lb)	51	409
Reserve (lb)	205	
Residual (lb)	31	5
Vent	25	
Pressurization		
He (lb)	22	
Tank (2) (lb)	62	
Valves & Lines (lb)	30	

in the main portion of the tank will be at saturated conditions; that is, most of it will be, at least to the extent of limited mixing caused by stratification during the gravity oriented phase. Small hydrogen tanks, located inside the larger ones, will be used to supply the Cryhocycle pump during this phase. The heat rates are low enough that the hydrogen can be pressurized with helium and supplied to the pump with a net positive suction pressure. A NPSP of 5 psia was assumed to be required. Approximately 2 minutes before liftoff the main tanks are locked up and the heat flux from them, until 2 min. into the ascent phase, causes a pressure rise to about 27 psia. The tanks can then be vented during the orbiter burn, while the liquids are still oriented. About 25 lb of hydrogen is vented. After venting, the tanks can be locked up and pressurized with 5 psia of helium and the feed switched from the small internal tanks to the main tanks. No additional venting is required and the pressure will slowly increase but stay below the 35 psia design value. A set of pressure and temperature transducers would have to be employed to maintain the helium at a 5 psia partial pressure.

During reentry, when the heat rates begin to increase, the small internal tanks would again have to be used to supply subcoooled hydrogen to the pumps. The oxygen is not pumped, so no specific NPSP is required. The method of supplying the oxygen is to pass whatever fluid enters the outlet through a preheat exchanger and an orifice to a hydrogen-oxygen heat exchanger. The hydrogen-oxygen heat exchanger is used to maintain both reactants at the same temperature for close mixture ratio control to the catalytic combustor. The oxygen tank is locked up at liftoff and the pressure permitted to increase throughout the flight. If no oxygen is used, the temperature will reach 150 psia and the tank will be vented. However, it is not necessary to separate the liquid from the vapor, since liquid venting will also lower the pressure; if the pressure has increased to 150 psia, more than sufficient oxygen is available for the remaining flight time. Normally the tank will function at about 100 psia.

The pressurant for the hydrogen is cold helium, stored at 4000 psia inside the main hydrogen tank. The helium is supplied to the hydrogen tank on demand from the logic system, which maintains a partial pressure of 5 psia. A continuous total pressure rise will take place with time; however, because of the continuous withdrawal of liquid, which causes hydrogen vaporization at the surface in proportion to the hydrogen partial pressure, the pressure does not rise fast and stays under the 35 psia design value.

The Cryohocycle subsystem weight is summarized in Table 3-14.

3.3.5.2.4 Freon Coolant Subsystem Loop. The same basic coolant loop, except for some modifications, is used with the Cryhocycle system as is used for the baseline system. Instead of circulating the Freon through the radiator, it is circulated through the Cryhocycle. The radiators are eliminated completely. The heat is transferred to the Cryhocycle machine via Freon-cryogen heat exchangers, which are part of the Cryhocycle subsystem weights. The baseline Freon loop weight was modified by elimination of the water sublimator, hydrogen heat exchanger, and some control valves associated with the radiator. The latter two functions are included in the Cryhocycle machine weights, as they are related to the Cryhocycle. The weight summary is shown in Table 3-15.

Table 3-15

FREON COOLANT SUBSYSTEM WEIGHT SUMMARY

GSE Heat Exchanger	(1)	25
Intercooler	(2)	80
Accumulators	(4)	20
Pump	(4)	25
Cold Plates & Heat Exchanger Valves & Reg.		130
Plumbing		75
		605
Fluid		400
		1,005

Table 3-14

CRYHOCYCLE SUBSYSTEM WEIGHT SUMMARY

	Sundstrand Data	Grumman Data
Cryhocycle (three expanders, associated heat exchangers and crossover modules)	910	528
Hydrogen Storage System (2)	568	614
Oxygen Storage System (1)	24	24
Helium Storage (2)	62	71
H_2 Supply valves and lines	74	74
O_2 Supply valves, lines, heat exchangers	27	27
He Supply valves and lines	30	30
Dry Weight	1705	1368
H_2	1461	1692
O_2	414	414
He	22	26
Fluids	1897	2132
Total	3602	3500
Batteries	220	220

3.3.5.2.5 <u>APU Subsystem</u>. The APU subsystem and power profile employed for the Cryhocycle is identical to the one employed for the baseline system. In this case the power capability of the Cryhocycle is not sufficient to deliver the large power demands of the hydraulic system; therefore, rather than force the Cryhocycle to be large enough for satisfying the reentry power functions, an entirely separate hydrazine supplied APU was assumed. The weight summary for this subsystem is listed under the baseline summary.

3.3.5.2.6 <u>Cooling During Reentry</u>. As explained for the baseline system, the only additional cooling required throughout the mission flight profile is during reentry. A total cooling requirement for that phase was postulated to be 363,000 Btu. A separate dedicated hydrogen cooling system was defined for the baseline to remove this amount of heat via hydrogen-Freon heat exchangers by venting hydrogen overboard. With the Cryhocycle operating, a portion of the cooling load can be absorbed by the machine. If it is run at its maximum heat absorbing capability (to 540°R), and the exhausting hydrogen is further heated in the APU subsystem, a total of 14,500 Btu/hr can be removed.

Since the Cryhocycle is already cooling the electrical load, the total additional heat that must be removed is 294,000 Btu. If the same assumptions are used here as were employed for the baseline, a total 170 lb of hydrogen (including a 10-percent reserve) must be added to the system. Since a storage system is already on board, it was assumed that this amount of hydrogen could be added to the tanks by making them slightly longer. The weight increments resulting from this are listed below:

Tank weight increment	19 lb
Pressurization weight increment	9
Heat exchanger and valves added	6
Total weight increment	34
Fluid	170
TOTAL	204 lb

3.3.5.2.7 Cryhocycle System Weight Summary. The various subsystem weights of the Cryhocycle system are summarized in Table 3-16.

Table 3-16

CRYHOCYCLE SYSTEM WEIGHT SUMMARY

	Sundstrand Data	Grumman Data
Cryhocycle machines	910 lb	528
Oxygen & Hydrogen storage	592	638
Helium storage	62	71
Supply & pressurization valves & plumbing	131	131
Batteries	220	220
APU	652	652
Freon coolant loop	605	605
Delta weight for cooling	34	34
Total Dry Weight	3206	2879
Fluids		
Hydrogen	1631	1862
Oxygen	414	414
Helium	22	26
Freon	400	400
Hydrazine	1206	1206
Total	6879 lb	6787

3.3.5.3 Baseline and Cryhocycle Systems Comparison. Figure 3-34 summarizes the total weights of the baseline and Cryhocycle systems. As can be seen, the Cryhocycle system weighs more than the baseline system. Two baseline system weights are shown, one employing subcritical tankage and the other employing supercritical tankage. If development risk is more important than weight, the system that employs the supercritical tankage would be preferred because more development experience exists on this system.

Two Cryhocycle system weight summaries are displayed, one based on the information developed in the Sundstrand Textbook "The Cryhocycle" and the other as developed from the NASA contracted "Shuttle Cryhocycle Study," performed by Grumman Aerospace Corp.

The baseline systems are between 285 lb and 534 lb lighter, depending on which of the two systems are compared.

The weight attributed to the basic Cryhocycle power generation and cryogens supply systems is approximately twice as heavy as the corresponding fuel cell systems. This results from the heavier Cryhocycle machinery, the larger amount of hydrogen, and the inefficient method of storing the voluminous hydrogen. The fact that no radiators are required for the Cryhocycle system gains back some advantage for it. The remaining systems are somewhat of equal weights.

The Cryhocycle system has the advantage of jettisoning a large portion of its gross weight in orbit. The hydrogen is expended during the orbital operations, so that upon landing the Cryhocycle system is lighter than the baseline system. The system landed and takeoff weights are shown in Table 3-17. If a constant wing loading at landing is assumed to be required, then a subsystem weight savings of 537 lb will permit an overall orbiter and subsystem weight savings of about 671 lb. However, if the subsystem liftoff weight increases by 534, the gross liftoff weight increases by 14,700 lb. This takes into account the orbiter weight savings obtained by the decrease in landing weight.

These various delta weights are shown in Fig. 3-34 for each of the systems as they are compared to the baseline, which was used as the reference. As can

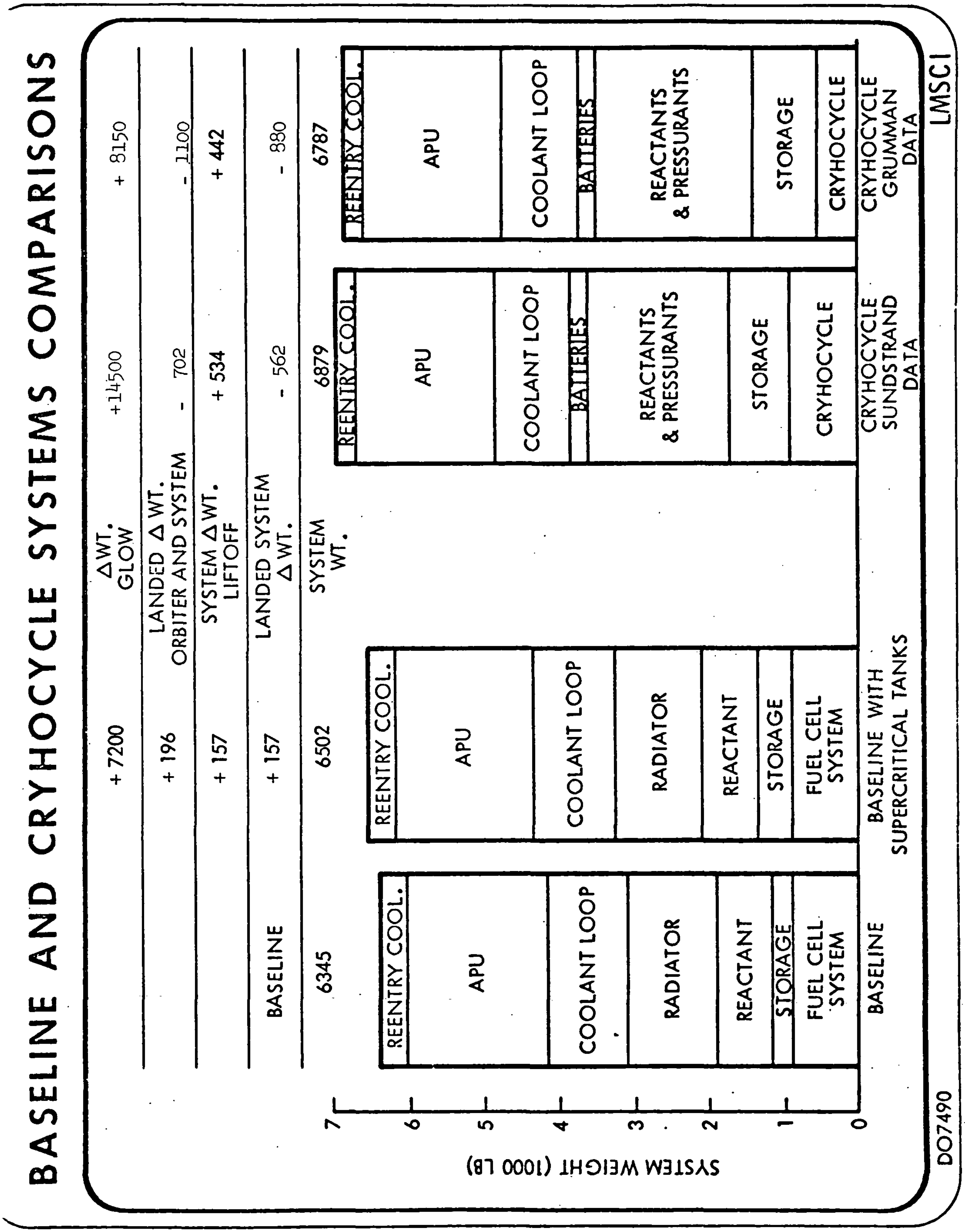


Fig. 3-34 Baseline and Cryhocycle Systems Comparisons

Table 3-17

SYSTEM LANDED AND TAKEOFF WEIGHTS

	Baseline System	Baseline System With Super critical Tanks	Cryhocycle System Sundstrand Data	Cryhocycle System Gruman Data
Gross system weight at Liftoff	6345	6502	6879	6787
Expendables				
H_2 Power	80	80	1430	1655
O_2 Power	642	642	409	409
N_2G_4 Power	1100	1100	1100	1100
H_2 cooling	191	191	170	170
Landed System Weight	4332	4489	3770	3452

be seen, the Cryhocycle system causes a large increase in the glow for both the Sundstrand and Grumman data. However, this weight increase is largely due to the solid rocket motor increase in size and is a relatively inexpensive item. If the subsystem costs are ignored, the overall program costs would be less for the Cryhocycle system.

As the mission time increase, the Cryhocycle system will become heavier as compared to the baseline system because of the poorer SHC and the added oxygen required for supplementing heat for a longer time. For the baseline system, no penalty would have to be added to the radiators because they can reject heat indefinitely.

Recent evaluations of the power requirements have shown that the electrical energy that must be generated could be as much as three times the value assumed for this study. In addition, the peak power might be as high as 23 kW during orbital operations. These peaks are relatively short. The increased requirements would create a proportional increase in the expandable reactants for both the baseline and Cryhocycle systems; that is, from 722 lb to 2166 lb of reactant for the baseline and from 1405 lb to 4215 lb for the Cryhocycle machine. The weight increase is larger for the Cryhocycle. The storage system weight will also increase more significantly for the Cryhocycles because large quantities of hydrogen are used, as contrasted to the baseline system which uses the dense oxygen.

The Cryhocycle machine weight will increase tremendously (not quite three times as much) if the same philosophy of having two standby units for fail operational-fail safe functions are maintained. Each of the three expanders and generators would have an output of 23 kW. If the peaks are assumed to be handled by batteries, the maximum power output would be about 18 kW, or nearly three times the value assumed for the study. Another option would be to operate two Cryhocycles at one half the design load for each. Four expanders would be required in this case, each having a power output of about 9 kW.

The fuel cells of the baseline system suffers from the same requirement; however, they can be operated at 100 percent overpower for short time periods and can therefore be designed for a lower average power output each.

The radiators would have to be larger; however, a growth of three times the size would not be expected. First, because the radiator weights used in this study are more than likely larger than are required for the baseline power profile and more heat could be rejected with them. Second, more of the fuel cell product water could be sublimed or evaporated to help reject the heat. Thus, for larger power profiles it is expected that the Cryhocycle would show more disadvantage than the baseline system.

3.3.6 EC/LSS and APU Cooling During Stowed Radiator Periods

3.3.6.1 Introduction. The subject of this section is the cooling of the ECS during periods when the Shuttle space radiators are stowed and therefore inoperative, and the cooling of the APU while it is operating. Past studies have shown that the cryogenically-stored hydrogen, which is heated and vented, provides an excellent means for cooling.

The determination of how much and to what degree hydrogen should be used for active cooling is very complicated. To clearly identify the advantages and disadvantages, alternate approaches must be investigated. Furthermore, the entire spectrum of the Shuttle flight profiles must be considered. This includes not only the ascent and reentry portions of the orbital flight but also the ferry flights and horizontal flight tests. These latter two must be considered because the development process formulated from their requirements could strongly influence the type of APU and ECS cooling designed into the vehicle. Also to be considered is the degree that one would want to remove or change equipment in the vehicle for a horizontal flight test to perform a nominal orbital flight. The amount and type of equipment change and additions to perform the ferry flight is also of importance, and the safety aspects of using hydrogen must be reviewed.

A matrix of various techniques that can be used to provide cooling for the various flight phases is shown in Table 3-18. This matrix includes many techniques that clearly are not weight optimum but may have potential from a schedule and operational point of view.

For the main orbital phase of the Space Shuttle mission, cooling loops and radiators will be used to reject heat from the ECS, fuel cells, cabin windows, etc., while the radiators are deployed. This has been clearly shown by various other Space Shuttle studies to be the best approach of those techniques listed in Table 3-18. However, during the ground hold, ascent, reentry, atmosphere flight, and postlanding ground cooling phases of the Space Shuttle mission, other heat rejection methods will be required for these systems. The horizontal flight testing and ferry flights will also require methods of heat rejection other than radiators. The object of this section, then, is to examine other possible methods of heat rejection during the stowed radiator phases of Space Shuttle flight.

Based upon various LMSC studies and Hamilton-Standard EC/LSS studies, the LH_2 vaporization and venting system of heat rejection appears to be the best for this application. This system is rather simple, has minimum weight, and can operate under all stowed radiator phases of flight and all gravity levels. An LH_2 vaporization and venting system would result in less dedicated liquid mass vented than for any other liquid chosen. All the heat exchangers could be designed for GH_2 heat transfer, so that boilers, liquid orientations, gravity levels, etc. would not be a major problem in a LH_2 heat rejection system.

The LH_2 system has some disadvantages, however, many of which are psychological. The possible problems associated with hydrogen loading, leaking, venting and purging, especially in the atmosphere, is a stumbling block to the acceptance of the LH_2 heat rejection system. All of these problems can be solved or minimized to a high probability of success, but there will always be reluctance to adopt this LH_2 system because of the explosive nature of hydrogen in the atmosphere. This would be especially true

Table 3-18

MATRIX OF COOLING TECHNIQUES FOR THE VARIOUS FLIGHT PHASES

FLIGHT PHASES \ COOLING TECHNIQUES	GSE	RADIATORS	EXPENDABLE VAPORS	• HYDROGEN	• WATER	• AMMONIA	• FREON	• CO_2	EXPENDABLE VAPOR COMBINATION	• WATER + AMMONIA	• WATER + HYDROGEN	• WATER + FREON	RAM AIR	MACHINES	• JET ENGINE BLEED AIR	• WATER WITH PUMP	• WATER-AIR/COMP. EXPANDER	• AIR CYCLE-PSUEDO CLOSED	• VAPOR CYCLE-CLOSED
ORBITAL FLIGHT																			
• GND HOLD	X																		
• ASCENT				X		X	X			2	2	2	3			X	X	X	4
• ORBIT		X																	
• DEORBIT/ENTRY				X	1	X	X			X	X	X				X	X	X	4
• ATMOSPHERIC/LANDING																			
• POST LANDING	X																		
FERRY FLIGHTS																			
• GND HOLD	X																		
• ATMOSPHERIC				X		X	X			2	2	2	3		X	X	X	X	4
• POST LANDING																			
HORIZONTAL FLIGHT TESTS																			
• GND HOLD	X																		
• ATMOSPHERIC				X		X	X			2	2	2	3		X	X	X	X	4
• POST LANDING	X																		

(1) Water can be used during atmospheric operation if supplemented with a pump.
(2) Nearly no water is used for these phases.
(3) Suitable for APU cooling but not for EC/LSS cooling unless used in conjunction with a machine.
(4) Cooled with water vapor outside of atmosphere and with air within the atmosphere.

during the early horizontal flight testing of the Space Shuttle, when its flight worthiness as an airplane is first being established and the inconviences associated with hydrogen use would probably be avoided.

The low temperature GH_2 flow into the heat exchangers will result in large Δ T's, and hence smaller area and mass heat exchangers would be required than would be possible with more conventional fluids. However, thermal stress problems may be worse in these heat exchangers. Although not much experience exists in the design and operation of these types of heat exchangers, studies and limited development work by AiResearch and Hamilton Standard have shown that it should be possible to design a trouble-free heat exchanger of this type. The major advantages of this LH_2 system are: (1) its lightweight, (2) it can be used for all the radiator-stowed phases of Space Shuttle flight, and (3) only a limited single development in the area of hydrogen heat exchangers is required.

It appears that all other heat rejection systems proposed for stowed radiator flight cannot be applied to all phases of that type of flight. For example, the air cycle cooling system could only be used in the lower atmosphere, whereas many liquid venting systems cannot be used in the lower atmosphere. It would be desirable to have one system, or at most a combination of two systems, that would require minimum penalties of weight, spacecraft changes, and operational procedures for the Space Shuttle. The following sections suggest alternate heat rejection systems that would be used in the stowed radiator phases of flight in case the LH_2 system is abandoned. The advantages and liabilities of these systems should be compared to the LH_2 system. Some of the concepts described, when considered from a power and weight point of view, clearly will not be appropriate to use for the Shuttle. However, since evaluations were made on these systems, it seemed prudent to include them among the potential concepts.

In the following sections the use of expendable fluids will be discussed, followed by a discussion of various water vapor, air cycle, and vapor compression machines. Following this is a section which discusses the

cooling aspects of the APUs. The various cooling techniques are then summarized by a narrative, followed by weight estimates of the most promising systems. Finally, the use of an engine bleed air cycle machine for cooling during ferry flights is given.

3.3.6.2 Heat Rejection by Expendable Evaporation System. Considering the non-cryogenic expendable evaporants, water has the highest latent heat of vaporization and hence would have the lowest boiloff rate for heat rejection. The major disadvantage of the water evaporation systems considered is the low vapor pressure of water at the EC/LSS coolant loop heat rejection temperature of 35 to 45°F. The object of this section is to examine other possible non-cryogenic liquid expendables, and to study the heat rejection capability of ammonia, which appears most promising.

To limit the rate of boiloff of liquid expendables, the latent heat of vaporization should be as high as possible. For vaporization to occur at sea level, the vapor pressure of the expendable liquid must be greater than 1 atmosphere at the temperature of about 35°F. Explosion, corrosion, and toxicity problems should not be too severe for the vented vapor.

Next to water, hydrogen fluoride has the highest latent heat of evaporation, of about 700 Btu/lb. However, this acid is very corrosive to metals, its vapor pressure is only about 1/2 atmosphere at the temperature of interest in this study, and would be very toxic; therefore, it is not considered further in this study.

Ammonia, with a latent heat of about 540 Btu/lb, has the next highest latent heat and also a vapor pressure > 1 atm at the EC/LSS coolant loop heat rejection temperature of 35 to 40°F. Ammonia is toxic to humans in small concentrations in the air. It is also corrosive to copper and copper alloys, but is not too corrosive to other metals. Except for these toxicity and corrosion problems, ammonia would be an excellent liquid evaporant. With a carefully designed system, both of these problems should be manageable.

A liquid with the next highest latent heat and a vapor pressure > 1 atm is methylamine, whose latent heat is about 360 Btu/lb. This alkaline liquid and vapor is highly flammable and corrosive to copper, brass, and aluminum, but not corrosive to steel. Its physiological effects on humans are similar to those of ammonia, but it is considered only moderately toxic. Its problems of combustion, toxicity and corrosion, especially towards aluminum do not make this evaporant look good.

The next liquids that have a vapor pressure >1 atm at the coolant loop heat rejection temperature of about 50°F, have latent heats in the range of 150 to 200 Btu/lb. These include liquids such as hydrogen sulfide, methyl chloride, and sulfur dioxide. The toxicity of these fluids, plus their low latent heat values, does not warrant further consideration of these evaporants. A large boiloff rate of these liquids would be required to obtain the desired heat rejection.

A desirable set of liquid evaporants, some with vapor pressures > 1 atm, are the Freons. The latent heat of these liquids are in the range of 60 to 100 Btu/lb, which is very low. The large boiloff rates and liquid loadings required for these evporants more than offset their desirable properties of no toxicity, corrosion, or flammability.

The high vapor pressure of carbon dioxide, CO_2, makes it a possible evaporant. As a liquid at 40°F, its latent heat is only about 100 Btu/lb. As a solid at $T \leq -70°F$, its latent heat of sublimation is about 240 Btu/lb. Because of the solid nature of CO_2 below the triple point, and its not too high a value of latent heat of sublimation, there seems to be little reason to consider CO_2 sublimation further.

The above survey of non-cryogenic evaporants with vapor pressures > 1 atmosphere shows ammonia to have the most potential for Space Shuttle heat rejection. Its high latent heat of about 500 Btu/lb offers significant weight savings of expendable liquid, but its undesirable properties of toxicity and corrosion with some metals will influence the design of an ammonia vaporization system.

Ammonia/water mixtures can be used as an expendable coolant. The higher latent heat of water plus the heat of solution of ammonia and water, should result in high effective latent heats of vaporization for aqua-ammonia solutions. However, after looking at enthalpy temperature plots for these mixtures and considering the methods of evporation that might be used, it does not appear as though ammonia/water mixtures offer any weight advantages compared to the evaporation of pure ammonia alone. The basic reason for this conclusion is the fact that the water fraction of the mixture cannot be effectively evaporated at heat rejection temperatures of 35 to 40°F. Since the water of the mixture cannot be effectively evaporated, the aqua-ammonia mixtures appear to require more expendable liquid mass than for pure ammonia evaporation alone. Hence, the venting of pure ammonia only will be considered as the best method of heat rejection in the remainder of this section.

Assuming an effective latent heat of vaporization of about 500 Btu/lb for pure ammonia, the required boiloff rate for the EC/LSS coolant loop for a heat rejection rate of 60,000 Btu/hr would be 120 lb/hr. This ammonia boiloff rate is about double the water boiloff rate of 60 lb/hr that would occur, assuming a latent heat of vaporization of 1000 Btu/hr for water. Also, this ammonia boiloff rate is about 3 times the LH_2 vaporization rate of 40 lb/hr that would result from the vaporization and heating of LH_2, with an effective latent heat of 1500 Btu/lb assumed. Hence, on an expendable mass basis, ammonia as an evaporant is about twice as heavy as water and about 3 times heavier than LH_2.

For the ascent and reentry modes of orbital flight, there is no question that some sort of expendable liquid evaporant will be required to provide EC/LSS heat rejection during these orbital mission phases when the radiators are not deployed. In the lower atmosphere, at altitudes less than 40,000 ft, the EC/LSS heat rejection can occur to expendable evaporants or to ram air. For the orbital flight mission, with less than one hour flight time in the lower atmosphere, a heat rejection system using an expendable evaporant only

will have the lightest weight. However, this conclusion is not necessarily true for the horizontal test or ferry flights in the lower atmosphere.

A 3,000-mile ferry flight, at M = 0.4 Mach number (300 mph) will require 10 hours of continuous flying time, assuming aerial refueling is used. All of the expendable evaporant will have to be loaded at take-off, which in the case of ammonia would be about 1200 lb and in the case of LH_2 would be about 400 lb or less of dedicated liquids. The take-off weight of these expendable evaporant systems would then be considerably more than the weight of the water evaporation/vapor compression refrigeration cycle that will be discussed in Section 3.3.6.8. This system would require no expendable liquid evaporants for flights in the lower atmosphere. Hence, for ferry and other lower atmosphere flight phases, an active refrigeration heat rejection system might be of lighter weight than some ammonia expendable evaporant system. Expendable evaporant systems such as ammonia or hydrogen have logistics problems concerned with supplying NH_3 or LH_2 to the Space Shuttle. The water evaporation/vapor compression refrigeration cycle does not have these supply problems.

3.3.6.3 Heat Rejection by Water Vaporization Systems. A water vaporization system could represent a reasonable approach, but studies to date have indicated that it cannot be used in the earth's atmosphere. Vaporization of Freon 22 or ammonia could be used during all mission phases, but the weight and/or toxicity problems become excessive. Active heat rejection systems such as the air cycle or vapor compression refrigeration systems have moderate weight, but are only useful in the lower atmosphere.

Because of the desirability of having only one heat rejection system for all mission phases, the availability of liquid water in the Space Shuttle, the potential low weight of a water vaporization system, and the safety aspects of this system, the water vaporization system should be explored in more detail. The objective of this section is to make a preliminary study of a water vaporization heat rejection system that could be used during all of the Space Shuttle mission phases.

For heat rejection from the Space Shuttle EC/LSS coolant loops, a water evaporator heat exchanger would probably have to operate in the $32°F < T_L < 45°F$ temperature range. This liquid water temperature corresponds to a vapor pressure of 0.09 psia $< p_v <$ 0.15 psia. A vacuum fore pump would be required to obtain these pressures in the earth's atmosphere. The water vapor densities over the above T_L range of temperatures would be $0.003 \text{ lb/ft}^3 < \rho_v < 0.0005 \text{ lb/ft}^3$.

With a latent heat of vaporization for water of about 1000 Btu/lb and a heat rejection rate of 60,000 Btu/hr, a water evaporation rate of about 60 lb/hr would be required to reject the heat from a EC/LSS water evaporator heat rejection system. For the SPU, a different set of heat exchangers operating at higher temperatures and heat rates would be used. Therefore, for the purposes of analyses, the two systems are kept separate.

The simples water evaporation system that could be devised is that shown in Figure 3-35. In the atmosphere, below an altitude $h \approx 110{,}000$ ft, a vacuum pump would be required to vaporize the water in the evaporator. The minimum possible pump work W_{ps} that would be required at sea level, $h = 0$ ft, would result from isentropic vapor compression.

$$W_{ps} = w\,(h_{2s} - h_1) \;=\; 12.36 \text{ hp for isentropic pumping.}$$

This pumping power requirement looks reasonable, but this is the minimum possible pump work requirement. The low vapor inlet density would result in a low efficiency vapor pump requiring a large volume flowrate. This pump would be large, with a low isentropic efficiency. Hence, the actual pumping power W_p would probably be more like 60 hp.

Because of the expected high pumping power requirement Wp and the probable large size and mass of this type of vapor pump, the above system does not appear too promising. However, the pumping power and compressor size might be reduced in the system shown in Fig. 3-36.

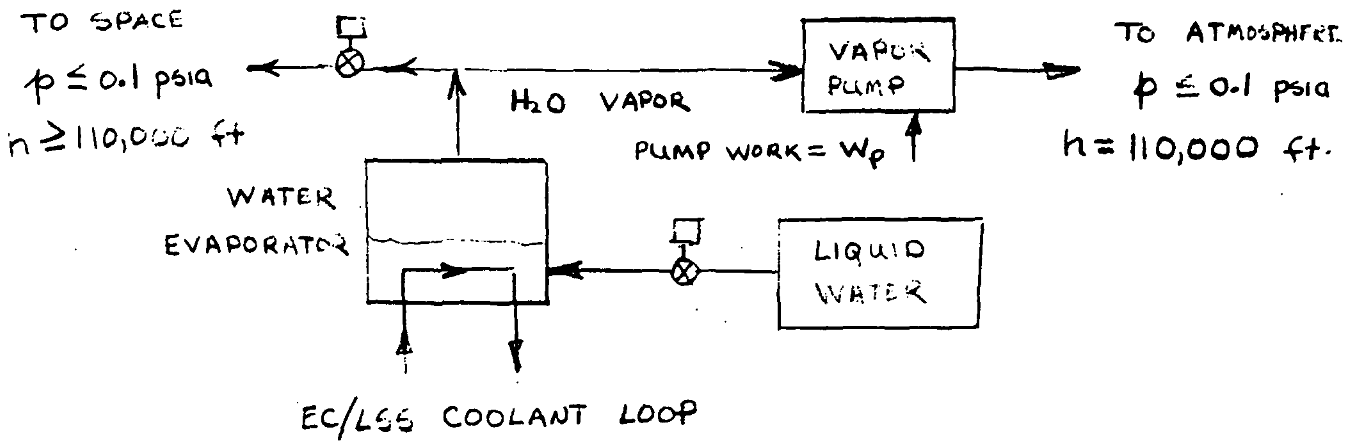


Fig. 3-35 Water Evaporation System

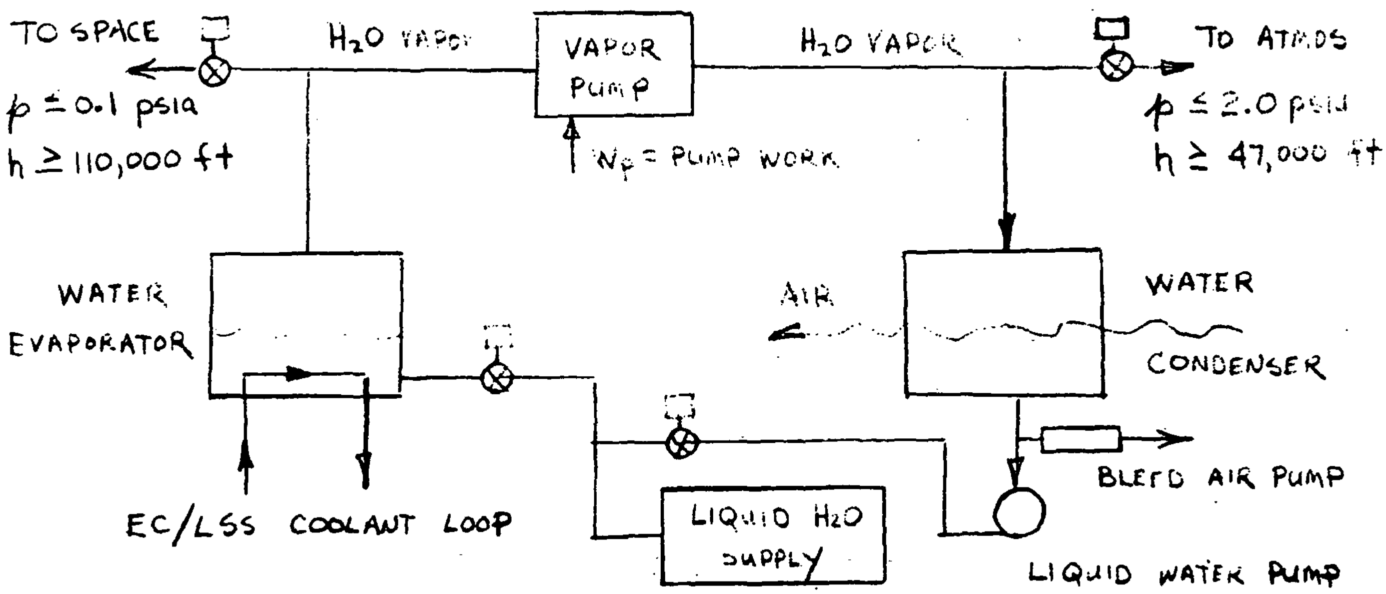


Fig. 3-36 Modified Water Evaporation System

At high altitude, $h \geq 110{,}000$ ft, this system would be a water evaporator exhausting directly to space. In the altitude range $110{,}000 \text{ ft} > h > 47{,}000$ ft, where the air pressure $0.1 \text{ psia} < p < 2.0$ psia exists, the vapor pump would operate and the water vapor would be exhausted to the atmosphere downstream of the vapor pump. For altitudes $h \leq 47{,}000$ ft and air pressures $p \geq 2.0$ psia, both water vapor vent valves would be closed, and the entire system would operate as a closed cycle, water vapor compression refrigeration cycle. At altitudes $h \leq 47{,}000$ ft, the air density is $\geq 1/6$ of that at sea level, with air temperatures $\leq$ sea level values, so that adequate air cooling of the water condenser should result.

At a vapor pressure $p_v = 2.0$ psia, then water condensing temperature $T_c \approx 126°F$ would result, so that it would always be possible to reject heat to the surrounding air. This system could operate over all Space Shuttle mission phases, although it has slightly more complexity than the typical vapor compression refrigeration cycle. Let us check the potential performance of this system, at the highest expected ground hold temperature.

The isentropic pumping power of the vapor compressor is estimated, based upon an outlet pressure $p_2 = 2.0$ psia. $W_{ps} = w\,(h_{2_s} - h) = 5.68$ hp for isentropic pumping.

The isentropic efficiency of this pump would be low, but not as low as one exhausting to atmospheric pressure at sea level. Hence, the pumping power requirement should be in the range 19 hp to 28 hp for this system. The pumping power would at least be cut in half by pumping into a condenser rather than into the atmosphere at sea level. The higher efficiency for this pump should result from the lower pressure ratio required.

However, this pump would also encounter the low vapor inlet pressure and density, so that a large pump with large volumetric flowrate would be required. Except for the lower pumping power, this pump will be about the same size and weight of the vapor pump that exhausts to atmospheric pressure.

The worst operating condition for the condenser would appear to be a hot day at sea level. With an air temperature of $T_A \approx 100°F$ and a condensing temperature $T_c = 126°F$, the coolant air temperature could only rise about $\Delta T = 20°F$ in flow through the condenser. Let us assume a condenser heat rejection of $q = 60{,}000$ Btu/hr, although it would be greater than this by the pumping work, and with the specific heat of air $C_p \approx 0.24$ Btu/lb°R, the required air flowrate w_A would be 12,500 lb/hr.

For an atmospheric air density of $\rho_A \approx 0.070$ lb/ft^3, the volumetric air flowrate would be 50 ft^3/sec hence a rather high air flowrate would be required through the condenser. With an air velocity through the condenser of $V_{AVE} \approx$ 10 ft/sec, an air flow frontal area of $A_F = 5.0$ ft^2 would be required. Hence, the condenser would probably have about the weight and size of an automobile radiator, with a fan to create the air flow and forced convection.

This condenser would require a bleed air pump to exhaust non-condensible gasses from the condenser, since air could flow into the system when the water vapor vent valves are opened. At altitudes $h \geq 5{,}000$ ft, air temperatures less than the freezing point of water will be encountered, especially above 15 to 20,000 ft. Hence, a design or control problem would exist in the condenser to keep the liquid water from freezing. This would probably be controlled by reducing the cooling air flow through the condenser during these flight conditions.

Except for the possibility of reducing the vapor pumping power by about one-half, compared to exhausting the water vapor to the atmosphere, this closed cycle condenser system appears to offer little advantage and only more complexity. However, below about 50,000 ft altitude it is a closed cycle, and the evaporant water would not have to be supplied and vented.

The open or closed water vapor compression refrigeration cycles considered above, which refrigerate down to near the freezing temperature of water, have some decided disadvantages. The low density of the water vapor produced in the evaporator requires large volume flowrate vapor pumps which operate over large pressure ratios. These pumps have low isentropic pumping efficiency and

require rather large diameter flow passages for the vapor flow. These vapor pumps, which must handle a large vapor volume flowrate, are rather large and heavy.

The above disadvantages are probably the major reasons why water vapor compression refrigeration cycles have not been used for aircraft ECS. If this system were considered further, the efficiency, size, weight, and pumping power input to the vapor pumps should first be examined.

The condenser for a closed cycle system should offer no problems, as it would only operate under one gravity conditions. The water evaporator must be capable of operation over the range of zero to reentry levels for these water evaporation systems.

3.3.6.4 Heat Rejection by a Water Evaporation/Air Cycle System

System 1. The water vaporization systems considered previously have the problems of pumping low density water vapor. In considering systems that are capable of performing for all mission phases, an evaporator is required for the upper atmosphere and space, whereas heat rejection to the atmosphere should be possible at altitudes $\leq$ 50,000 ft, with some sort of refrigeration cycle. To develop a single simple system that would utilize the most readily available working fluids and expendables, it would appear that a combination water evaporation/air cycle system might be possible, and at least should be considered as a possible heat rejection system. The most obvious water evaporator/air cycle system would be that shown in Fig. 3-37.

For altitudes above $h \approx$ 110,000 ft, the water evaporator could yield temperatures as low as T_{OUT} = 35 - 40°F for the EC/LSS coolant loop. To obtain a significant rate of evaporation, and to be able to exhaust the water vapor to space with the attendant pressure drops, the water evaporator might only be effective for altitudes $h \geq$ 150,000 ft.

The open cycle, air cycle refrigeration system shown downstream of the water evaporator in the EC/LSS coolant loop could be used for cooling in the lower

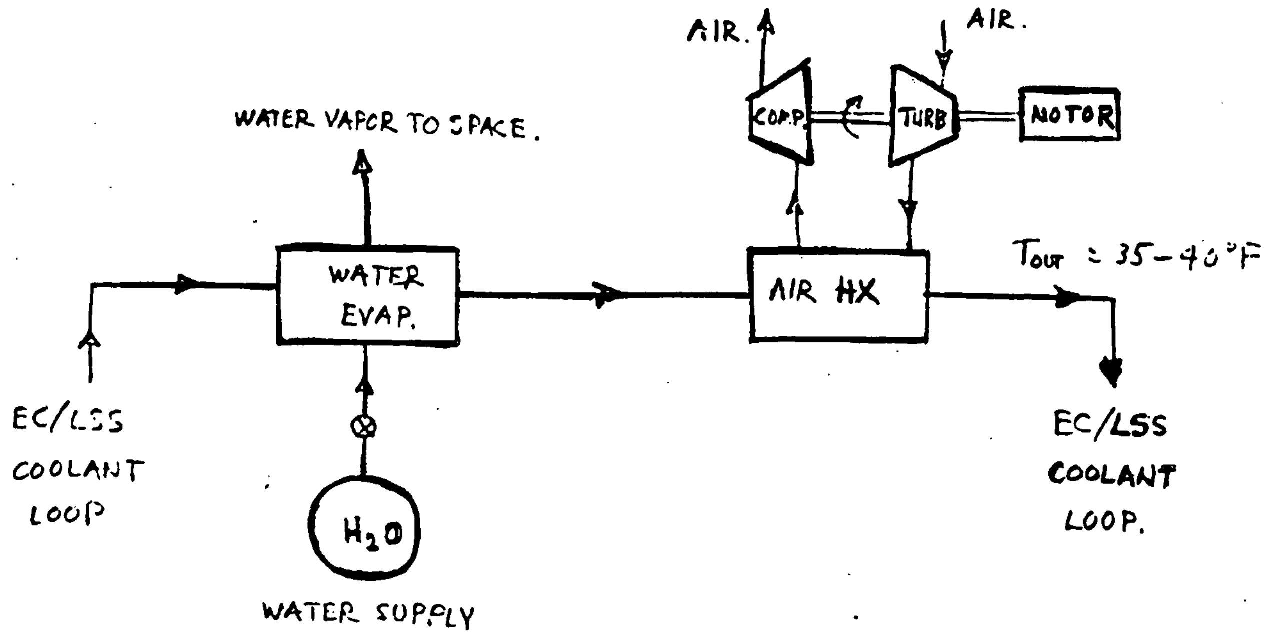


Fig. 3-37 Water Evaporation/Air Cycle System 1

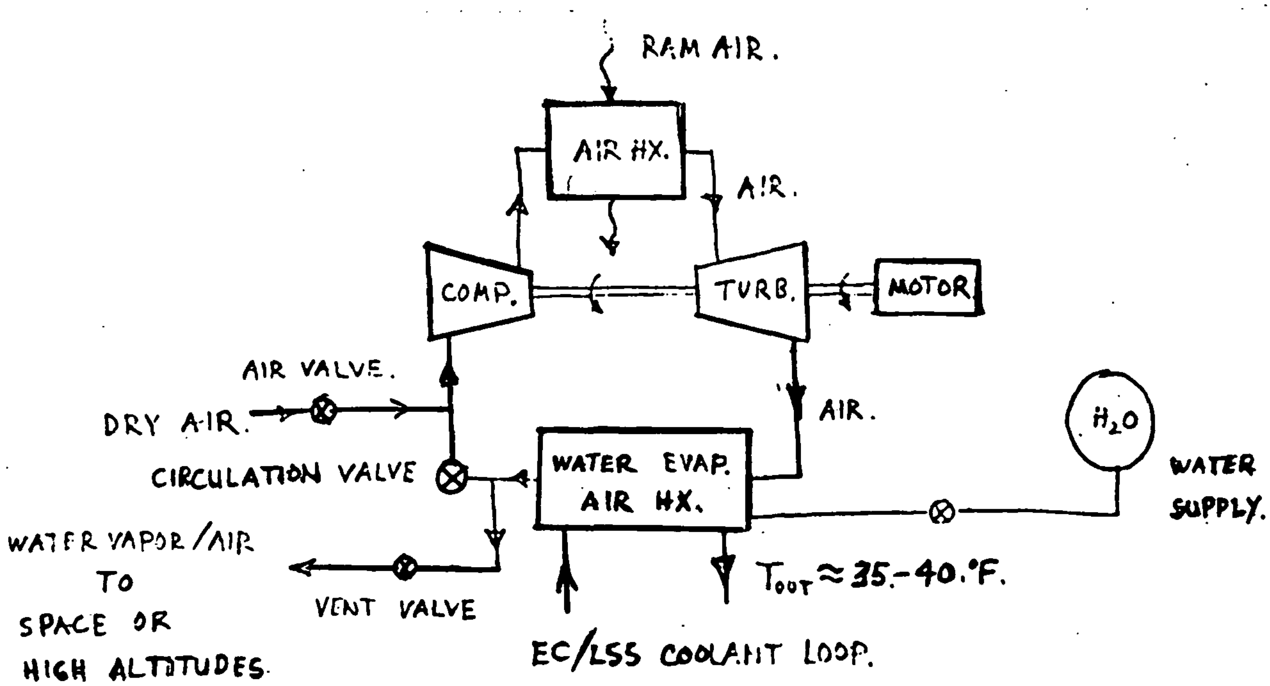


Fig. 3-38 Water Evaporation/Air Cycle System 2

atmosphere. Conventional air cycle refrigeration machines can be used up to altitudes of $h \approx 50,000$ to 60,000 ft. This open cycle machine would use atmospheric air as the working fluid, hence no expandable evaporants would have to be supplied for atmospheric curise or ferry flights, as well as possible ground hold conditions. If this system is to be used for all possible mission phases, a bleed air refrigeration system should not be used, since various reentry missions are planned to not have the jet engines on the orbiter. As shown in the sketch, a motor could be used to drive the air cycle machine. This system would probably required about a 30 hp motor, which should result in an air cycle flowrate ω_A of about a few thousand lb/hr of air flow.

The major disadvantage of the system is that it will not provide cooling in the altitude range 50,000 ft $\leq h \leq$ 150,000 ft. During ascent, the Space Shuttle spends a time period of about one minute in this altitude range, whereas on reentry descent the time period in this altitude range would be about 10 minutes. Since EC/LSS cooling should be provided during these times, especially for the longer descent time period of about 10 minutes, the system shown in Fig. 3-37 is not adequate for all mission phases.

System 2. To provide cooling during the ascent and descent time periods discussed above, the water evaporator/air cycle system shown in Fig. 3-38 appears feasible to use. Except for some added control valves, this system has the same components as that shown in Fig. 3-37. A detailed analysis of this system has not been made, but a description is given to show what should be possible with the combination water evaporator/air cycle machine.

For altitudes $h \geq$ 150,000 ft, the system would act as an expendable water evaporator system, with the air cycle motor off, the liquid water supply valve open, and the vent valve open to vent the water vapor to space. The vent valve would control the vapor pressure, and hence the temperature in the water evaporator. The operation would be that of the typical expendable water evaporator previously used on spacecraft.

For low altitude operation in the atmosphere, say for altitude $h \leq$ 40,000 to 50,000 ft, this refrigeration system would operate as a closed cycle, air

cycle system. The water supply valve, vent valve, and air valve would be closed, and the circulation valve would be open, with the motor driving the air cycle machine. Heat rejection would occur from this closed air cycle to ram air or compartment air through the upper air heat exchanger.

Air cooling of the EC/LSS coolant loop would be accomplished by flowing the cold turbine outlet air through the vapor passages of the water evaporator. Hence, the EC/LSS coolant heat exchanger would be designed as a combined water evaporator/air heat exchanger. This combined water evaporator/air heat exchanger will probably dictate the closed air cycle machine described above for low altitude, and especially sea level operation. An open air cycle system would result in dust or dirt contamination of the water evaporator spray and/or wicking systems that would be required for zero gravity operation. Hence, for low altitude operation of $h \leq 40{,}000$ ft, the air cycle machine should be operated as a closed cycle machine to reduce contamination of the water evaporator/air heat exchanger. For low altitude operation in the atmosphere, then, this system would operate as a closed air cycle refrigeration machine, without any water vaporization or expendable water supply required.

For operation in the altitude range $40{,}000 \text{ ft} \leq h \leq 150{,}000$ ft, the system would operate as a combined water evaporator/open air cycle machine. From Fig. 3-39 it can be seen that for altitude $h \geq 40{,}000$ ft, the dew or freezing point temperatures T_F are $T_F \leq 360°R \approx \leq -100°F$, which is much less than the atmospheric air temperature $T_A \geq 390°R \approx \geq -70°F$, for altitudes in the range $40{,}000 \text{ ft} \leq h \leq 150{,}000$ ft. Hence, the atmospheric air at this altitude is very dry, with a dew or freezing point temp $T_F \leq -100°F$. Water evaporation into this low pressure air at a temperature $T_L \approx 35$ to $40°F$ should be rather simple to induce in the water evaporator.

For the altitude range $40{,}000 \text{ ft} \leq h \leq 150{,}000$ ft, the operation of the system would be as described below. The air cycle motor would be on, the water supply valve, air valve, and vent valve would be open, and the circulation valve would be closed. Hence, the system would operate as an open air cycle machine,

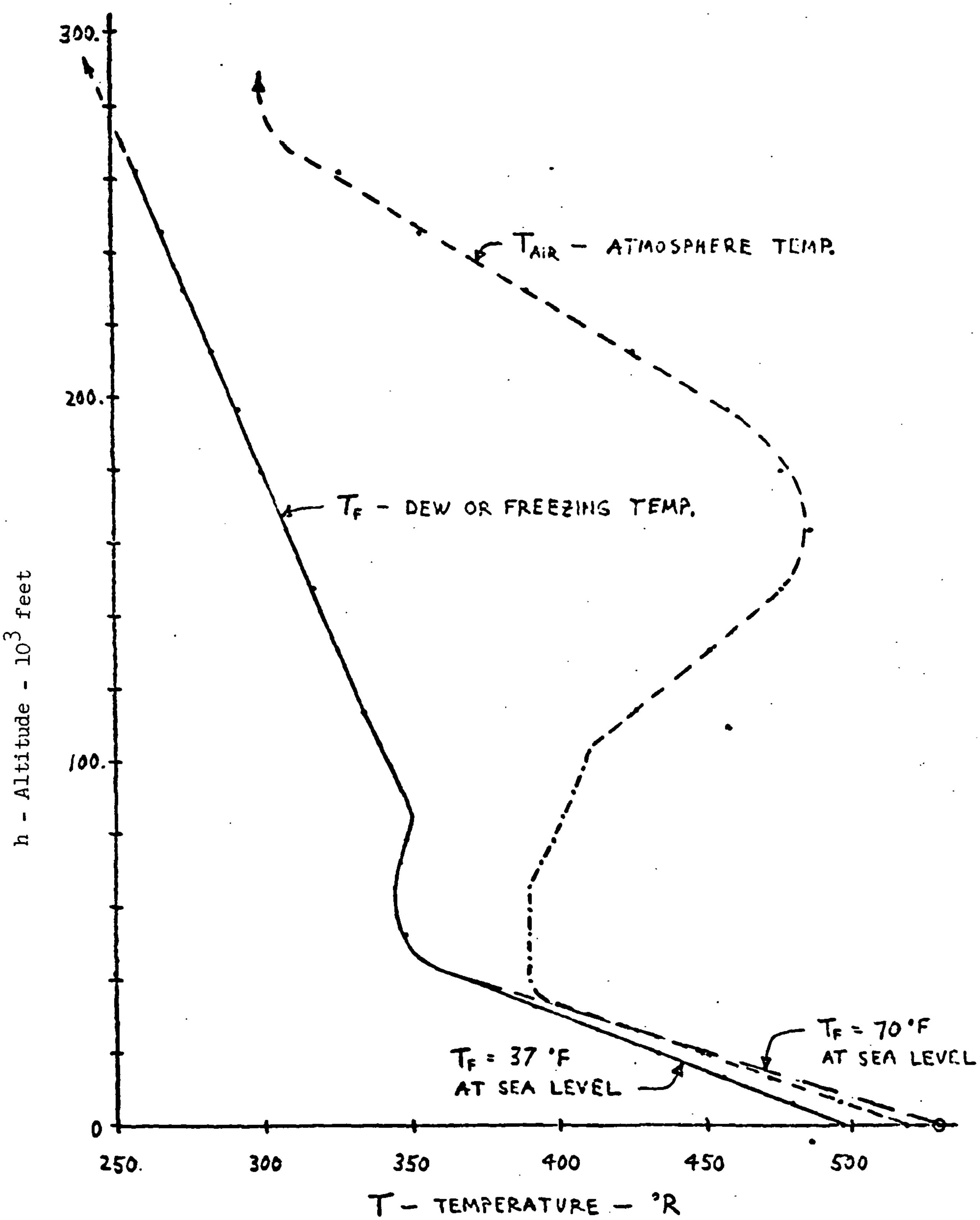


Fig. 3-39 Atmospheric Temperatures and Dew Points versus Altitude

blowing low density and cooled air through the water evaporator vapor channels. This system would be a combined water evaporator/open air cycle system for the altitude range specified at the top of this paragraph.

At the higher altitudes $h \leq 150{,}000$ ft, this system would operate more as a water evaporator. Assuming an EC/LSS coolant heat rejection rate of $q \approx 60{,}000$ Btu/hr, a water evaporation rate of $\omega \approx 60$ lb/hr would be required to provide this cooling. From low pressure psychrometric charts, air humidities of ≥ 0.1 lb H_2O/lb air should be expected to leave the evaporator for altitudes $h \geq 100{,}000$ ft; hence, the air cycle unit would only have to supply ≤ 600 lb/hr of air flow through the water evaporator at altitudes $h \geq 100{,}000$ ft. The air cycle unit, at these high altitudes, would tend to blow the water vapor through the evaporator with rather small flow rates of dry air. For altitudes $h \geq 100{,}000$ ft, most of the heat rejection from the EC/LSS loop would be due to expendable water evaporation, with only a small amount of air cycle refrigeration. The small air cycle flowrate would serve more to blow the water vapor from the evaporator, rather than provide refrigeration at the lower altitudes.

For operation in the altitude range $100{,}000 \text{ ft} \geq h \geq 40{,}000$ ft, transition would occur from mostly water evaporation cooling to mostly air cycle refrigeration at lower altitudes near $h \approx 50{,}000$ ft. As the altitude is lowered in this range, less water evaporation and more air flow would occur through the system. Hence, at an altitude of $h \approx 40{,}000$ ft, sufficient air flow would be available in the cycle to obtain all the refrigeration from the air flow. At this time, the air circulation valve could be opened, and the vent valve, air valve, and water supply valve could be closed, after which the unit would operate as a closed air cycle refrigeration system. The above description is a rough idea of how this combined water evaporator/air cycle refrigerator would work during the reentry phase.

During reentry dry atmospheric air would always be available in the altitude range $150{,}000 \text{ ft} \geq h \geq 40{,}000$ ft. However, during ascent, air taken from the compartment during this altitude range would probably be of high humidity,

being ground hold air in the Space Shuttle. Ram air could be used during this ascent time period, but a cooling heat exchanger would probably be required. This ascent time period in the altitude range $40{,}000 \text{ ft} \leq h \leq 150{,}000 \text{ ft}$ only lasts for about $\Delta t \approx 1.0$ min., and the heat rejection $\Delta Q \approx 1000$ Btu that would have to be rejected from the EC/LSS coolant loop during this short Δt might be handled by transient capacity effects, if the proper supply of dry cooling air was not available during the open air cycle operation of the cooling system during ascent.

The above description gives a rough idea of the proposed operation of the combined water evaporator/air cycle refrigeration machine. The system appears to be possible, but more detailed analysis of the open cycle, water evaporation phase of operation would be required to demonstrate its possible performance. This analysis would have to include the effects of water evaporation into forced convection, low density, and dry air streams. The system would appear to reject heat over all proposed mission phases for the Space Shuttle.

The advantages of this combined water evaporator/air cycle system are that it uses two very available working fluids, air, and water. This system could make use of the saem EC/LSS water evaporator that seems to be planned for orbital excess heat rejection, except that this evaporator will be heavier because it must reject heat at a large rate. An air cycle or vapor compression refrigeration cycle would probably be used for heat rejection in the lower atmosphere, and air cycle machines are usually lighter. A motor would be required to drive the air cycle machine, with about the same, or slightly more, horsepower than that required for a vapor compression cycle. The motor allows the air cycle to be closed and separated from the jet engine air bleed, which will not be present for all mission phases. Electrical or hydraulic power should always be available on the Space Shuttle for this system. This combined system for all mission phases should be lighter than all other comparable systems, except for the LH_2 evaporator heat rejector.

The disadvantages of this system would appear to be centered about the analysis and development costs required to build the combined water evaporator/air heat exchanger for the EC/LSS coolant loop. The operation and control of this system would not appear to afford any major problems. This system could be used from start to finish on the Shuttle.

3.3.6.5 Heat Rejection by a Water Evaporator/Vapor Cycle System. This system could use a water evaporation/vapor compression cycle combination, which is sketched in Fig. 3-40.

The EC/LSS coolant loop runs from right to left along the top of the above sketch. A zero gravity water boiler will reject heat from the coolant loop at altitudes h ≥ 150,000 ft in the atmosphere or in space. At lower altitudes, heat rejection would occur through the evaporator/coolant heat exchanger of the refrigeration cycle.

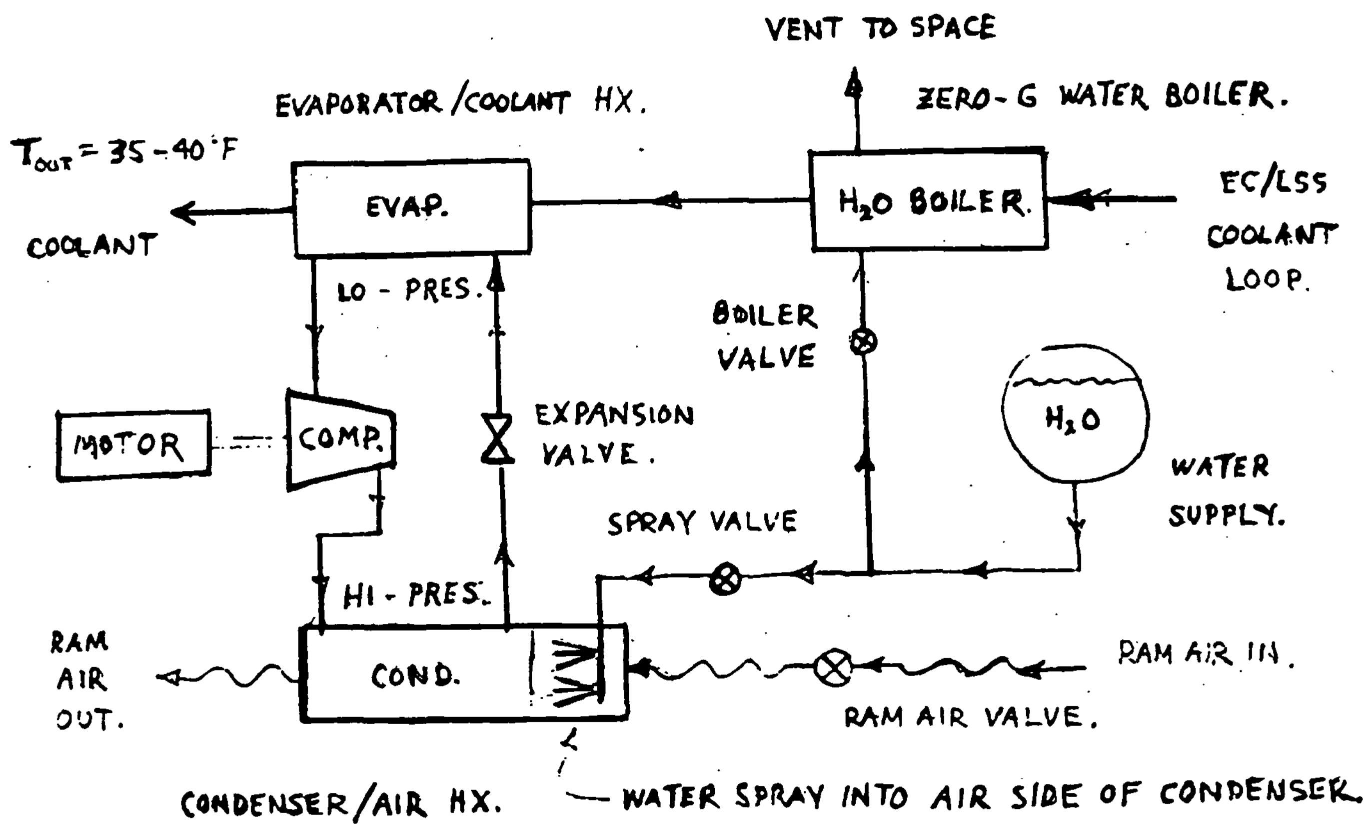


Fig. 3-40 Water Evaporation/Vapor Cycle

The refrigeration cycle would then reject heat through the condenser/air heat exchanger shown at the bottom of the sketch. At altitudes h ≤ (about 50,000 ft), the condenser would reject heat to the ram air flow shown. At altitudes h ≥ 50,000 ft, water would be sprayed into the air side of the condenser. This water would evaporate/boil on the air flow surfaces of the condenser, and the vapor would exit through the ram air outlet. Hence, in the altitude range of 50,000 ft ≤ h ≤ 150,000 ft, the condenser/air heat exchanger would reject heat as a water boiler.

A vapor compression refrigeration cycle is shown in the sketch. It would also be possible to use a closed air cycle refrigeration system for this application. For an air cycle machine, the expansion valve would be replaced by an air turbine, which would help drive the compressor. The air cycle could be hermetically sealed, or it could be filled just prior to the start of each operation by a supply of air from the Space Shuttle LSS.

The vapor compression refrigeration cycle should be the logical choice for this system. The motor power should be less for a vapor compression system than for a closed air cycle of the same refrigeration capacity. Although the weight of vapor compression cycles are usually greater than bleed air refrigeration systems, they should not be heavier than a comparable closed air cycle system. The closed air cycle would also have motor weight, plus the weight of two heavy air heat exchangers corresponding to the evaporator and condenser shown on the sketch. In the vapor compression cycle, the evaporator/coolant heat exchanger should be lightweight. A detailed trade study would probably show a decided advantage of the vapor compression refrigeration cycle for this application.

The data of Fig. 3-41 was plotted to help demonstrate how this system will work. The curve T_B shows the water boiling temperature as a function of altituded h in the atmosphere, with the water vapor pressure being assumed equal to the static air pressure. The T_{As} curve shows the ascent ram air stagnation temperature as a function of altitude h for a nominal ascent trajectory. The times shown next to the symbols of this curve are the

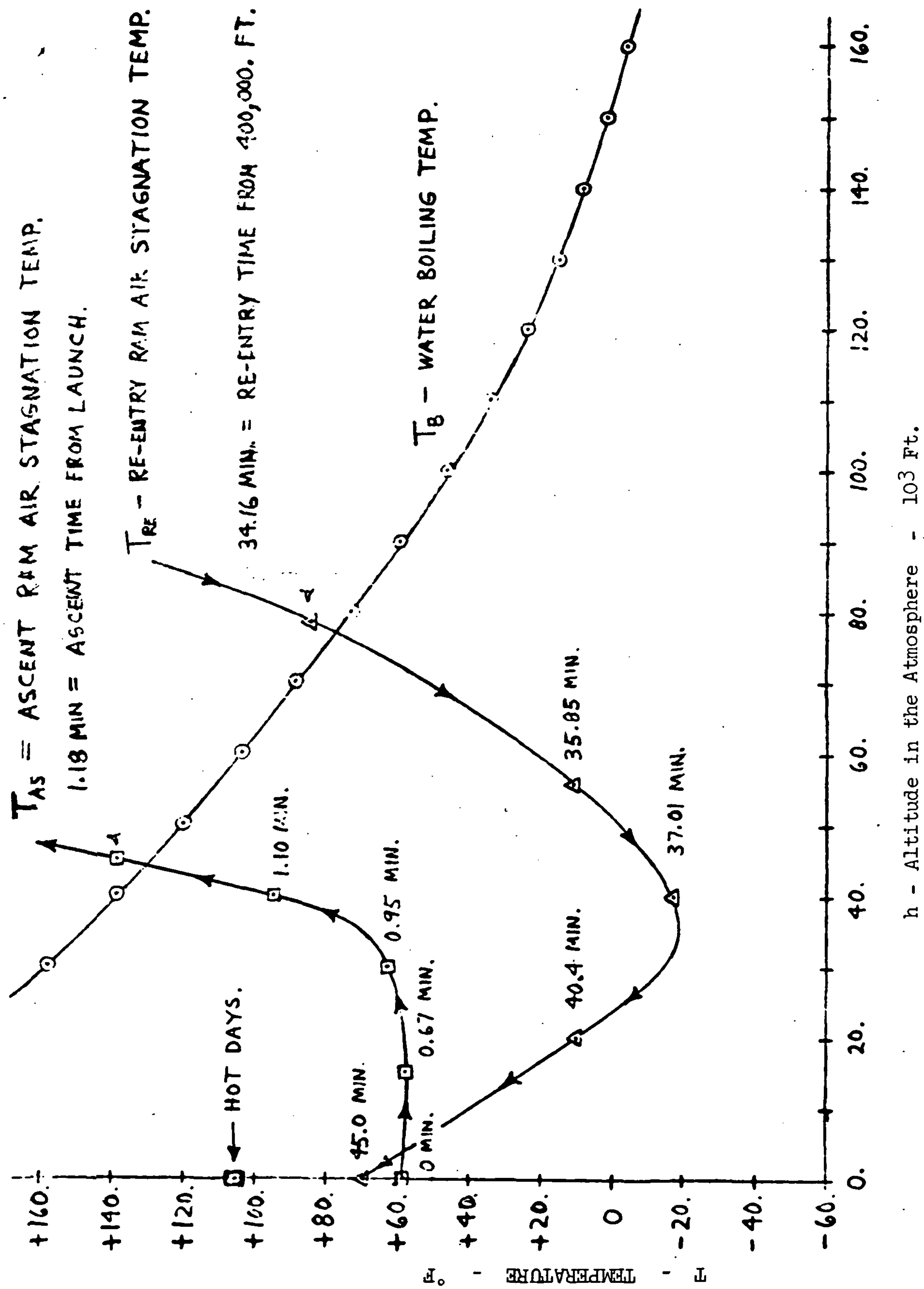


Fig. 3-41 Heat Rejection System Temperatures

ascent times from launch in minutes for those points. The curve T_{RE} represents the reentry ram air stagnation temperature as a function of altitude h for a nominal 1100 NM crossrange reentry trajectory. The time in minutes shown next to the data points of this curve are the reentry time that has elapsed from the start of reentry at 400,000 ft altitude.

Based upon Fig. 3-41, the condenser/ram air heat exchanger combined with a water spray should be able to reject heat from the refrigeration cycle at a condenser temperature $T_c \leq 130°F$ during both the ascent and reentry flight phases. With the evaporator temperature $T_E \approx 35$ to 40°F, the coefficient of performance (COP) for this refrigeration cycle should be similar to that of a typical household refrigeration system. Assuming a coolant heat rejection rate in the evaporation of $q_E = 60{,}000$ Btu/hr and COP = 2.0 for the refrigeration cycle, then the compressor pump work W_p would be 11.78 hp and the heat rejection from the condenser q_c would be 90,000 Btu/hr. If water spray evaporation only is being used to cool the condenser, then a water flowrate $\omega \approx 90$ lb/hr would be required, assuming the latent heat of vaporization of water is $h_\lambda \approx 1000$ Btu/lb. This condenser water evaporation rate requirement is about 1/2 more than the $\omega \approx 60$ lb/hr of water evaporation that would be required in the zero gravity water boiler to obtain the same coolant loop heat rejection. This water spray evaporation on the condenser would only last for short periods during the ascent and reentry phases of flight.

For ram air cooling of the condenser (no water), the system would be designed for hot day, sea level operation, with $T_A = 105°F$ and $T_c = 130°F$. The ram air flowrate required through the condenser would be 15,000 lb/hr or 3300 cfm air at sea level. This ram air flow and volume rate appears reasonable for the condenser. For reentry at 40,000 ft altitude, from Fig. 3-41 the quantity $(T_c - T_A) \geq 100°F$, more than 4 times the hot day sea level value of $(T_C - T_A)$. Hence, one-fourth the air flow rate at that altitude would be able to reject the heat rate $q_c = 90{,}000$ Btu/hr from the condenser. Since the air density at 40,000 ft is about one-fourth that at sea level, one-fourth of the sea level air flowrate would result from the same volume rate of $W_A = 3300$ cfm at 40,000 ft altitude. For reentry, cruise, or ferry flights,

only ram air cooling of the condenser will be required up to an altitude of 40,000 to 50,000 ft. At altitudes above this level, water spray cooling will be required to augment or substitute for the ram air cooling. The pumping power, ram air coolant rates, and water spray evaporation rates look good for this cycle.

The operation of the heat rejection system, sketched in Fig. 3-40, during the ascent and reentry phases of flight is considered in the following paragraphs. The discussions will make frequent references to the data on Fig. 3-41.

Ascent Flight. At the time of launch, the refrigeration cycle would be on, with the ram air valve open, but the water spray and boiler valves would be closed. This operation would continue to an altitude $h \approx 45{,}000$ ft, when the ram air valve would be closed because the ascent ram air stagnation temperature would be too high above this altitude. The water spray valve to the condenser would be opened just prior to closing the ram air valve, so that water boiling would occur on the condenser for $h \geq 45{,}000$ ft. At an altitude $h \geq 150{,}000$ ft., the boiler valve to the zero-g water boiler could be opened, and heat rejection could be assumed by this component. The refrigeration cycle could then be stopped, and the water spray valve closed. The time from launch required to reach $h \approx 150{,}000$ ft altitude is 2.17 min., so that only ≈ 1.0 min. of water evaporation on the condenser would be required on ascent before the space water boiler could be used to reject heat from the coolant loop.

Because of the higher stagnation air temperature during ascent, pure ram air cooling of the condenser will not be adequate in the altitude range 20,000 ft $\leq h \leq$ 45,000 ft, above which water evaporation cooling of the condenser will be adequate. During ascent, the Space Shuttle spends only $\Delta t \approx 0.4$ min. $\approx$ 24 sec in this altitude range, and less than the required rate of ram air cooling during this short time will only increase the condenser temperature somewhat. Transient thermal capacity effects should be adequate until water spray boiling would commence on the condenser.

Reentry Flight. At altitudes $h \geq 150{,}000$ ft, the space zero-g water boiler would be rejecting heat from the coolant loop, with the boiler valve open, the spray valve and ram air valves closed, and the refrigeration cycle off. At $h \approx 150{,}000$ ft, the refrigeration cycle would be turned on, and the spray valve opened. At an altitude $h \approx 110{,}000$ ft, the zero-g water boiler would stop evaporation, the boiler valve would be closed, and heat rejection would occur by water spray evaporation in the condenser. At an altitude of $h \approx 75{,}000$ ft, the ram air valve could be opened, since the reentry ram air stagnation temperature would be below the water boiling temperature T_B at that altitude. At an altitude $h \approx 40{,}000$ to $50{,}000$ ft, the water spray valve could be closed, and the condenser would be cooled by ram air flow only below this altitude level. For reentry flights, about 10 minutes of time is spent in the altitude range $150{,}000 \text{ ft} \geq h \geq 40{,}000$ ft, and this is the time period when spray water evaporation cooling of the condenser will be required.

Ferry and Horizontal Flights. For altitudes $h \leq 40{,}000$ ft, heat rejection from the EC/LSS coolant loop is provided by the refrigeration system rejecting heat to ram air. Hence, for the initial horizontal test flights and ferry flights, no water evaporation or water supply is required for these operations. Therefore, the initial horizontal test flights could be performed without the zero-g water boiler installed in the Space Shuttle. The water boilers and spray system would only be required for the first orbital flight.

The combined water evaporation/vapor compression refrigeration cycle, sketched on Fig. 3-40, will reject heat from the EC/LSS coolant loop over all Space Shuttle mission flight phases. This system has the following advantages, and is considered to be much better than the combined water evaporation/air cycle system shown in Fig. 3-38.

a. Either a closed air cycle or a vapor compression refrigeration cycle can be used in this system.

b. Both the zero-g water boiler and the spray water evaporation on the condenser employ water boiling phenomena, where the vapor pressure is greater than the total pressure of the atmosphere.

c. There is no humidity or dry air requirement for the operation of this system. The system will perform in completely humid air.

d. Spray water boiling on the air side of the condenser will only occur under gravity conditions. There is no need for wicking materials in this boiler, and hence no contamination will occur with ram air flow through this condenser.

e. It should be simple to design this refrigeration cycle condenser/ ram air and water spray evaporation heat exchanger.

f. This system provides adequate heat rejection for all flight phases with a condenser temperature $T_c \leq 130°F$. Only for a short time span of $\Delta t \approx 0.4$ min. will the condenser heat rejection be less than required, but transient thermal capacity effects will handle this.

g. Only 10 to 15 hp should be required for the refrigeration cycle, for a EC/LSS heat rejection of q = 60,000 Btu/hr. About 90 lb/hr of water spray and/or 3300 cfm of ram air is required to cool the condenser.

The cooling techniques discussed previously were all directed toward providing cooling for the EC/LSS, where the temperatures had to be 35 to 40°F, at least in part of the system, for humidity control purposes. The higher heat loads imposed by the APU and hydraulics were not included because of higher temperatures at which these systems can operate permits, in addition to hydrogen cooling, water boilers and ram air cooling can be used more easily and compactly. These systems can be used for all flight phases including

horizontal flight if the APU is used. However, there are other considerations as indicated below that lead one to conclude that the APUs should not be used for ferry and horizontal flight tests.

3.3.6.6 Heat Rejection from the APU. The auxiliary Power Units (APU) will provide power for the hydraulic systems and supplementary electrical power during ascent and descent. During this period of time, cooling must be supplied. The characteristics of the cooling system may depend upon whether or not the APUs are used during the ferry flights. There are two possible methods of providing hydraulic power (i.e., pressure) for use by the aero-surface controls during the ferry mission: (1) operate the APUs throughout the flight, and (2) provide hydraulic power from hydraulic pumps installed on the turbo-jet ferry engines.

Of the two possible methods, the use of the on-board APU pumping system is the lightest and requires the least conversion time because additional equipment does not have to be added to the vehicle. However, there are several disadvantages if the APUs are used.

(a) Hazardous liquids must be loaded on board the vehicle. The APUs that drive the pumping system are powered by hydrazine, which must be loaded into the vehicle tank. In addition, if the hydraulic oil temperature is maintained by evaporating expendable fluids, then additional fluids must be loaded into the vehicle to supply the heat exchangers.

(b) Tank capacities for both the hydrazine and expendable fluids are designed for orbital missions and will limit ferry flight duration unless auxiliary tanks are installed.

(c) APU operating costs are high. The APUs have approximately a 500-hour operational life at which time they must be removed from the vehicle, the catalyst replaced, and a maintenance inspection performed. Therefore, each ferry flight will use up to one percent of the operating life between overhauls. In addition, four operating APUs will consume approximately 1000 lbs hydrazine per hour, which also increases the operating cost.

Installing hydraulic pumps, alternators, and ram air oil coolers on the ferry turbo-jet engine eliminate the problems associated with operating the APU during a ferry mission. In addition, the hydraulic pumps are interchangeable with the pumps installed on the APUs, and the existing turbo-jet engine have drive pads which are capable of providing the power required to drive the hydraulic pump and alternator needed for the ferry mission. The disadvantages of providing hydraulic power from the turbo-jet engine drive pads are (1) the weight and cost of bolt-on equipment (hydraulic pumps, alternator, and ram air cooler), and (2) the time to install this equipment on the engines.

After considering both alternative hydraulic power sources, it has been decided that the cost of the additional bolt-on equipment that must be installed on the turbo-jet engine is outweighed b the higher cost of operating the APU during the ferry mission; therefore, the hydraulic power is assumed to be provided by the turbo-jet engines and the associate bolt-on equipment. Cooling for the APU operating during ferry flight, therefore, need not be considered. Hydraulic fluid cooling can be accomplished by an oil-air exchanger that can be installed as part of the jet engine ferry package.

The heat rejection from the APU coolant loops will have rather large heat rejection rates, in the range 200,000 to 300,000 Btu/hr. This heat rate results from cooling the hydraulic oil, lube oil, electrical alternator, and turbine shields. The APUs are scheduled to be used during a one-third hour ascent period and during a 1-1/2 hour reentry period in connection with orbital flights. Hence, the APUs heat rejection system must operate from a low gravity earth orbit down to sea level in the atmosphere. During horizontal test and ferry flights, with air breathing engine hydraulic power, the APUs might be used only during short-term emergency periods.

The best temperature for the hydraulic oil during APU operation is about 180°F. For short time periods, oil temperatures as high as 250°F could be tolerated, but maximum steady oil temperatures nearer 200°F would be desirable. A hydraulic oil temperature $T \geq -40°F$ must be held before APU start-up so that the oil can be pumped. During APU operation, minimum hydraulic oil temperatures $T \geq 0°F$ should always results.

The APU coolant loop must reject heat to an expendable liquid evaporant or possibly to ram air at lower altitudes in the earth's atmosphere. To hold the hydraulic oil temperature near 180°F during APU operation, expendable liquid

evaporants such as LH_2 or ammonia could be used from space down to sea level. If water were used as the liquid evaporant, then an evaporator could be used down to an altitude of about 30,000 ft, with ram air cooling required at lower altitudes down to sea level. All of these evaporators will have to operate from zero g to a few g's acceleration, with variable directions to the acceleration vector.

The APU coolant loop could be used in conjunction with, or kept separate from, the EC/LSS coolant loop. The APU coolant loop should be a higher temperature, higher heat rejection rate coolant loop than that of the EC/LSS. The APU coolant loop need only operate when the APUs are on; hence, there seems to be little reason to circulate this fluid through the EC/LSS coolant loop with radiator, etc. However, if expendable hydrogen is used with the EC/LSS system, the heat capacity still remaining can be used for APU heat rejection.

Liquid water would be a good coolant for this APU loop, if operating temperatures in the range $32°F \leq T \leq 212°F$ are expected. However, during orbital conditions with the APU off, this H_2O coolant would probably freeze. If water could not be used as the APU coolant, then ethylene glycol/water mixtures might be used. These mixtures have good heat transfer properties, and low freezing points of about -40°F can be obtained. A more standard coolant, such as the Freons, coolants, etc., could be used, but these coolants have poor heat transfer properties.

The oil/coolant heat exchangers on the APU might create design and/or weight penalty problems. These heat exchangers will probably be located downstream of the oil pumps, and hence must be designed for high pressure oil flows. The hydraulic oil pressure will be as high as 3000 psi for the APU, with lube oil pressures of about 500 psi expected. Both the lube oil and the hydraulic oil are high Prandtl number fluids, and as such have very poor heat transfer characteristics. Hence, high pressure oil passages with large heat transfer areas (fins) will be required for the oil/coolant heat exchangers. If the coolant also has poor heat transfer properties, the heat exchangers will

probably be difficult to design and fabricate, with thermal and pressure stress problems, as well as being rather heavy. A coolant with good heat transfer properties should be helpful in the design and performance of these oil/coolant heat exchangers, but the oil side should govern the heat exchanger design.

Based upon the above considerations, one integrated coolant loop could be used to cool all the APUs, or a coolant loop could be provided for each APU, as shown in Fig. 3-42.

This APU coolant loop is rather simple, with the coolant flow created by the coolant pump when the APU is in operation. The temperature control of this loop would probably be designed to maintain an outlet temperature range for the hydraulic oil leaving the heat exchanger. This temperature could be controlled by throttling the rate of evaporation or ram air cooling in the evaporant heat exchanger, and by controlling the coolant flowrate. Parallel flow loops, or more complex controls, would be required if it is also necessary to control lube oil and/or alternator temperatures.

An upper limit estimate of the expendable liquid evaporant mass requirements to cool the APU follows. A maximum heat rejection rate of 300,000 Btu/hr will be assumed for all the APUs in operation. The APUs will run for an ascent time period of 0.3335 hr and a descent time period of 1.55 hr, which includes one possible fly-around. The total APU operation time per orbital mission would be 1.88 hr, maximum. An upper limit estimate of the total heat rejection load is 566,000 Btu.

Assuming that only liquid evaporants are used to obtain this cooling, the following expendable liquid masses would be required for maximum heat rejection.
LH_2 evaporant, where $h_\lambda \approx 2080$ Btu/hr ($T = 150°F$)

Mass = 272 lb

For H_2O (water) evaporant, where $h_\lambda \approx 1000$ Btu/lb :

Mass = 566 lb (too high a temperature in lower atmosphere)

For NH_3 (ammonia) evaporant, where $h_\lambda \approx 500$ Btu/lb:

Mass = 1132 lb

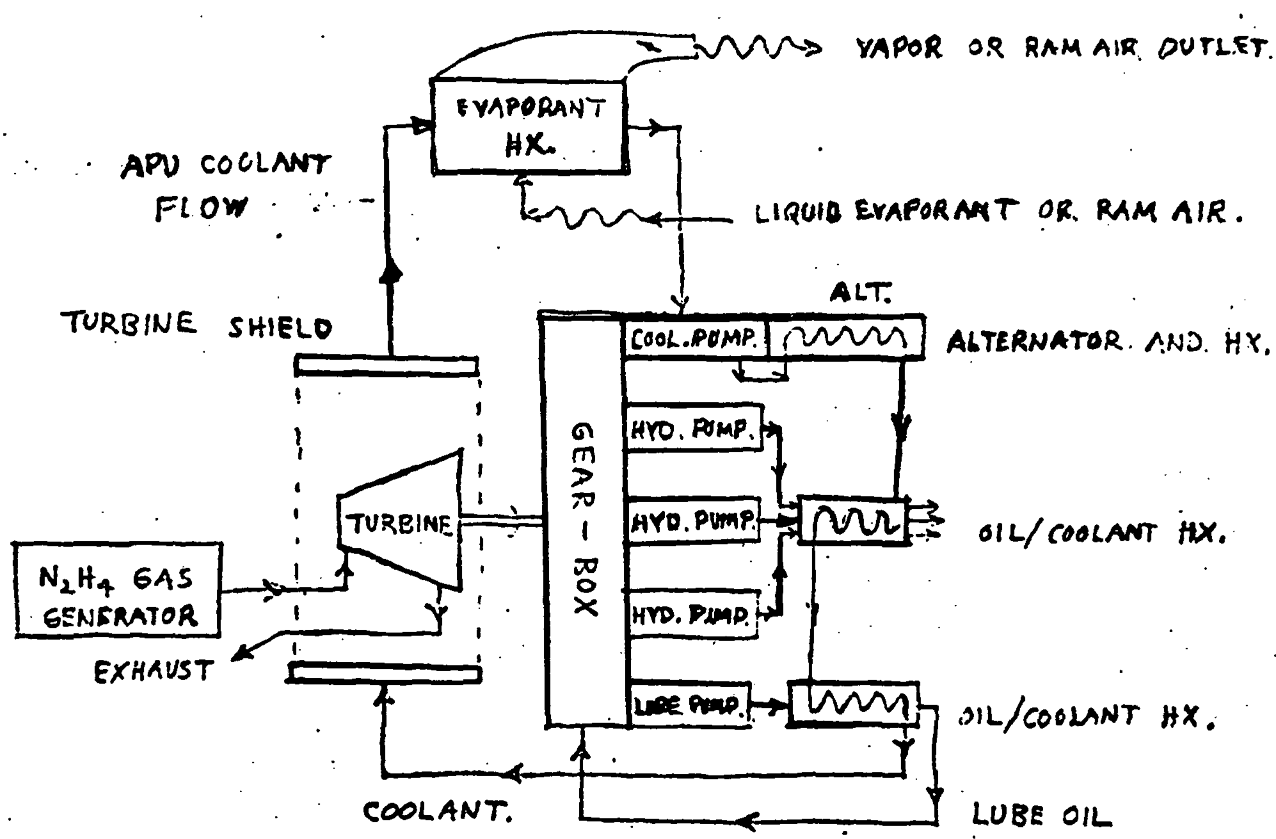


Fig. 3-42 Typical APU Coolant Loop

Hence, considerable dedicated liquid mass would be required to cool the APU under maximum heat rejection. If some of the APUs could be shut down during the APU system operation, which would decrease the system heat rejection, then a weight savings would result for the expendable evaporants, as well as for the hydrazine fuel.

If a combination water evaporator/ram air cooler is used for the APU system, the water requirement for an orbital mission would be reduced. Assuming that ram air cooling would be used for ground hold and altitudes less than 30,000 ft, the time periods for water evaporation cooling would be 1.5 hours.

The maximum heat load that must be rejected by water evaporation with a combined water evaporation/ram air cooler would be 450,000 Btu, and the mass of water evaporant would be 450 lb.

This combination cooler would require about 100 lb less water than a pure water evaporator, but the expendable water would still be heavier than that required for a pure hydrogen vaporization cooler.

A combination water evaporator/ram air cooler appears to be useful for both the APU and EC/LSS coolant loops. It is likely that the heat exchangers for these systems are not available, but the technology should be available to build them. They would have to operate from zero-g to a few g accelerations, with variable directions to the acceleration vector. Because air would flow through the cooling core in the lower atmosphere, wicking material should not be used for low-g water distribution because of contamination problems. A water spray would appear to be a better way to deliver the water to the evaporating surface. A design that incorporates swirling vapor flow through and out of the heat exchanger core should reduce the loss of liquid water droplets to an acceptable level.

The core of this heat exchanger would undoubtedly be a finned tube design, similar to an automobile radiator, with coolant flow inside the tubes. The finned surfaces would be the ram air and/or water evaporation heat transfer surface. This heat exchanger could be used as an APU cooler or a heat

rejector for a refrigeration cycle to cool the EC/LSS coolant loop. This cooler can always reject heat in the temperature range 100°F to 150°F, using the two common coolants — water and/or air.

Since the APU system has considerable thermal capacity, the possibility of running the APU under transient warming conditions should be examined. Dry weight of the APU and near thermal structure is estimated to be about 1000 lb, with another 1000 lb of oils, both hydraulic and lube. The thermal capacity of the system would be 700 Btu/°R.

If the system average temperature were to rise 100°R from the start of operation of the APU, the capacitance heat absorption would be 70,000 Btu. This is more than 12 percent of the total maximum heat rejection expected for an orbital mission. Hence, the thermal capacitance of the APU should reduce the maximum expected boiloff of liquid evaporants by more than 12 percent. Since the thermal capacity of the APU system will be fixed, the capacity will have a larger influence on reducing the amount of liquid evaporants required, if the total heat rejection from the APU can be lower than the reduction of the heat rejection alone would suggest.

A rough estimate can also be made of the time required for the APU system temperature to rise an average of 100°R. For the maximum expected heat rejection rate of 300,000 Btu/hr, the time period required to supply the capacitance heat absorbtion would be about 14 minutes. This time period is less than the minimum expected APU operating time during ascent of 20 minutes. If the heat rejection rate was lower or the time period of ascent operation was shorter, it might be possible to approach the condition of ascent APU operation without active cooling. However, even if this condition occurred for an average APU system temperature rise of 100°F, hot spots would probably exist in the APU during warmup, because the thermal energy would not be evenly distributed. Hence, it does not appear possible at this time to run the APU without active cooling. This active cooling requirement should be especially true for APU turbines, which should warmup in a time period of seconds after APU start-up. Hence, the APU turbine shields would require active cooling

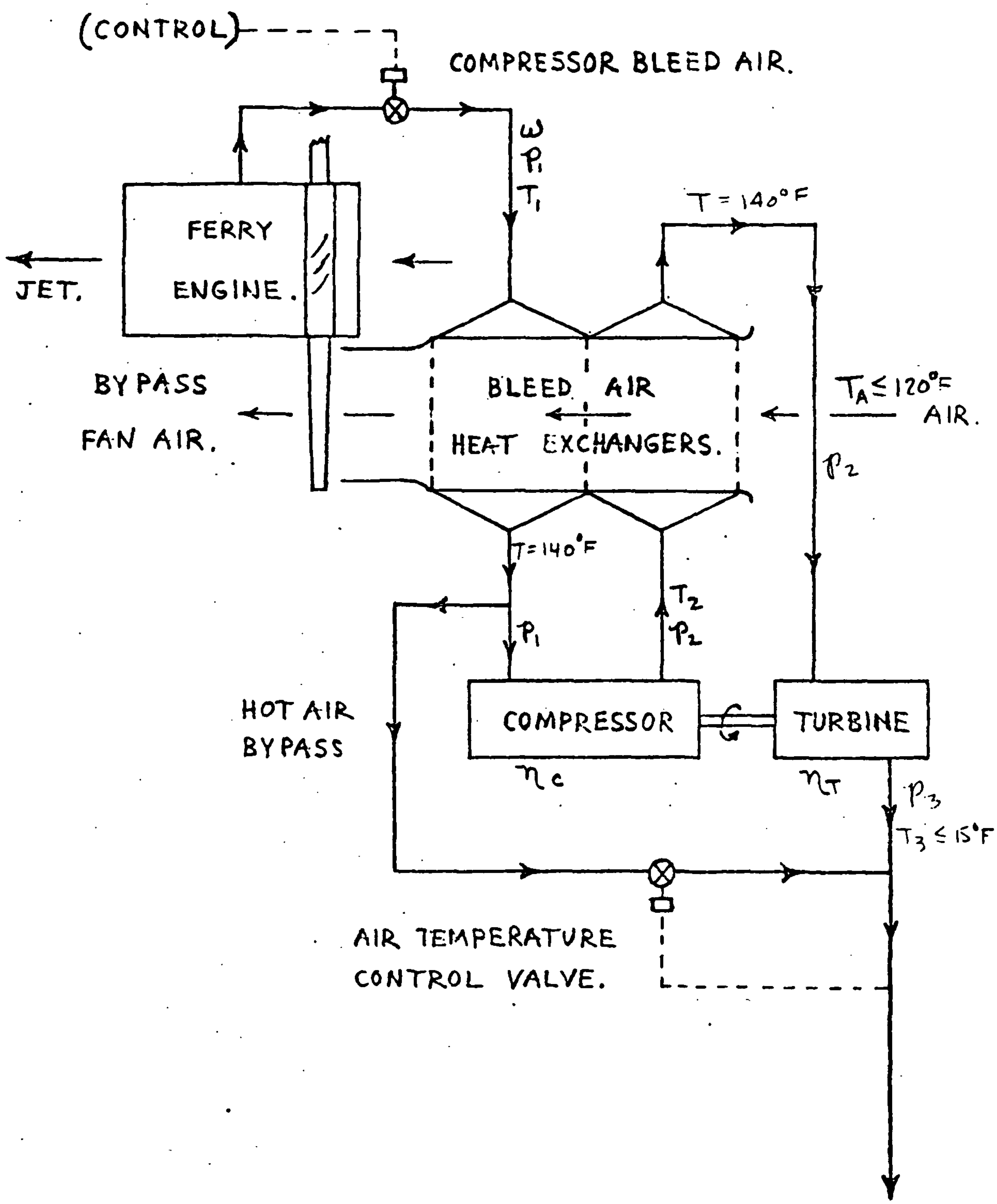
shortly after APU start-up. A rather quick, but somewhat slower, cooling requirement would probably also be needed for the APU alternators from system start-up.

3.3.6.7 Air Cycle EC/LSS Heat Rejection System for Atmospheric Flights. For atmospheric flight missions, especially those of long duration, air cooling systems might offer weight and logistics advantages when compared to expendable liquid evaporant systems.

Because of the expected reluctance to use hydrogen heat rejection for the horizontal test and ferry flights, it was decided to study air cycle cooling using jet engine air bleed for these flight phases. The air cycle heat rejection system could be mounted on, or in association with, the Space Shuttle air breathing ferry engines and hence would offer essentially no weight penalty for the orbital mission. The object of this study is to consider the design and problems of an air cycle heat rejection system for atmospheric flights. This air cycle system need only produce cooling for the EC/LSS loop, since the APUs will not be operated during atmospheric flights.

A wide range of air cycle cooling systems are possible for the Space Shuttle EC/LSS during atmospheric flight. For this study, it was assumed that the same orbital coolant loops, cabin ventilation, and contaminant removal systems are used in the EC/LSS for atmospheric flights. Hence, this air cycle system would only supply cooling to the EC/LSS loops, but not ventilation to the crew cabin. This will eliminate crew cabin contamination and pressure regulation systems that would exist with open cabin ventilation.

3.3.6.7.1 Jet Engine Bleed Requirements. An estimate of the bleed air flow rate required to perform cooling of the Space Shuttle ECS was made. Heat rejection rates in the range 20, 40, 60 and 80 K Btu/hr from the ECS are considered. An open cycle, air bleed refrigeration system, similar to that shown in Fig. 3-43, is assumed. Engine bleed air temperatures and pressures are assumed based on typical shuttle airbreathing engines.



Cold Air for
EC/LSS Cooling
$T_4 \leq 80$ to 100°F
$D_4 = P_3 = 14.7$ psig

Fig. 3-43 Schematic Diagram of Air Cycle Machine for a Ferry Engine

The use of bleed air obtained from the jet engines is subject to bleed air conditions (temperature and pressure) and engine bleed rate limits. The jet engine performance penalties that accrue due to the use of this bleed air must also be given consideration. The jet engine compressor bleed characteristics presented in Table 3-19 are considered to be representative of candidate airbreathing engines to be used for Space Shuttle ferry missions. Two flight conditions are noted, both assumed to be operating at maximum engine rpm on a standard temperature day. Engine bleed rate limits presented may not be exercised simultaneously. The combined bleed rate from the interstage and high-pressure bleed ports must not exceed the maximum rate assigned to either one. Fan bleed is independent of the other two bleed ports. During engine operation with bleed air extraction, the engine fuel control will reset engine rpm to correspond to the commanded engine power setting, provided that the turbine inlet temperature limit is not exceeded. Thus, thrust reduction due to compressor/fan bleed is minimized at the expense of engine S.F.C. On hot days the turbine inlet temperature limit will be encountered under some operating conditions that will prevent the engine fuel control from maintaining a constant engine rpm. Engine thrust penalties will be greater than shown in Table 3-19. Engine size adjustments to maintain a specified thrust level may be made using a thrust/weight ratio of 6.9 and an engine frontal area change directly proportional to the change in thrust. No engine length change is estimated to be required.

3.3.6.7.2 Expander and Cooler. The worst operating condition for the air bleed refrigeration system appears to be a hot day at sea level. A simple schematic diagram of the system, shown in Fig. 3-43, indicates temperatures and pressures shown for this most severe operating condition.

In this analysis, the performance of the heat exchanger is not analyzed and it is assumed that the outlet temperature of the bleed air from the primary and secondary heat exchanter is T = 140°F = 600°R. It is assumed that the turbine and compressor will have efficiencies η_T and η_c, respectively. It is also assumed that the highest possible outlet temperature T_4 of the bleed air from the ECS heat exchanger will be 20°F less than ECS coolant loop inlet

TABLE 3-19

JET ENGINE COMPRESSOR BLEED CHARACTERISTC

ALTITUDE FT	MACH NO.	BLEED TEMP (°R)	BLEED PRESS. (PSIA)	MAX. BLEED (LB/SEC)	ΔF LB	ΔSFC LB/LB
BY-PASS FAN BLEED						
Sea Level	0.0	713°	35.7	2.3	-189	+.006
10,000	0.4	685°	27.9	1.8	-120	+.007
COMPRESSOR INTERSTAGE BLEED						
Sea Level	0.0	1119°	135	2.5	-75	+.011
10,000	0.4	1105°	107	2.0	-53	+.016
HI-PRESS. COMPRESSOR BLEED						
Sea Level	0.0	1530°	418	2.5	-57	+.017
10,000	0.4	1517°	334	2.0	-30	+.021

temperature of T_{in} = 100°F to 120°F max. The bleed air inlet temperature T_3 to the ECS heat exchanger should also be more than 20°F less than the ECS coolant outlet temperature of T_{out} = 35°F. Fan power for the ram air and other minor systems have been neglected in the above model.

Assuming that the air bleed inlet temperature to both the compressor and turbine is T = 140°F - 60°R, one can set the compressor power equal to the turbine power, and hence find the turbine outlet temperature, T_3, as a function of the compressor inlet pressure, p_1:

$$W_c = \omega \left(h_2 - h\right)$$

$$W_c = \frac{\omega\, C_p T}{\eta_c} \left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right]$$

$$W_T = \omega \left(h - h_3\right) = \eta_T \;\omega\, C_p\, T \left(1 - \frac{T_{3_s}}{T}\right)$$

$$W_T = \eta_T\, \omega\, C_p T \left[1 - \left(\frac{p_3}{p_2}\right)^{\frac{k-1}{k}}\right]$$

Now, set $W_c = W_T$, with temperatures T = 600°R assumed, so that:

$$\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1 = \eta_c\, \eta_T \left[1 - \left(\frac{p_3}{p_2}\right)^{\frac{k-1}{k}}\right]$$

Since p_3 = 1 ATM and p_1 = bleed air pressure, which is probably known, then $p_3/p_2 = \left(p_3/p_1\right)\left(p_1/p_2\right)$, where p_3/p_1 should be a known quantity.

$$\left(p_2/p_1\right)^{\frac{k-1}{k}} = \frac{\left(1 + \eta_c\, \eta_t\right) \pm \sqrt{\left(1+\eta_c\eta_T\right)^2 - 4\, \eta_c\eta_T \left(p_3/p_1\right)^{\frac{k-1}{k}}}}{2} = r_p$$

$$p_2/p_1 = \left(r_p\right)^{\frac{k}{k-1}} \quad \text{or} \quad p_2 = \left(\frac{p_2}{p_1}\right) p_1 = p_1 \left(r_p\right)^{\frac{k}{k-1}}$$

The outlet temperature T_3 from the turbine can now be found from $W_c = W_T$:

$$\frac{T_3}{T} = 1 - \frac{1}{\eta_c}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right], \text{ where } T = 140°\text{F} = 600°\text{R}.$$

The air bleed flowrate, w, can then be found from the ECS heat exchanger rate q and the assumed air bleed outlet temperature, T_4.

$q = \omega C_p\ (T_4 - T_3)$, where $T_4 \approx 80°F$ to $100°F$, depending on coolant inlet temperature $\omega = \dfrac{q}{C_p(T_4-T_3)}$, assume minimum $T_4 = 80°F = 540°R$.

Now, compute the air bleed flowrate required for the following operating conditions. The air bleed outlet temperature T = 600°R from the primary and secondary heat exchangers, $\eta_c = 0.60$ and $\eta_T = 0.70$ for the compressor and turbine efficiencies. For the hot sea level condition, the outlet pressure $p_3 = 14.7$ psia and the bleed air pressure p_1, will depend on where the air is bled from. We shall assume perfect gas air, with $C_p = 0.24$ Btu/lb°R and k = 1.40 for the following solutions.

By-Pass Fan Bleed: (Sea Level), High Efficiencies. For seal level operation, $p_1 = 35.7$ psia from Table 3-19. The solution to the quadradic results in two positive roots, + .288 and +1.132.

The root less than 1.0 indicates that the compressor would act as a turbine, and this solution is of no interest. Only the positive root is of concern

and $\dfrac{p_2}{p_1} = (r_p)^{\frac{k}{k-1}} = (1.132)^{3.5} = 1.543$, $p_2 = 55.1$ psia.

The outlet temperature, T_3, from the turbine is then 8°F. Hence, it appears that temperature T_3 will be low enough.

Now, compute the bleed air flowrate, ω, as a function of the heat rejection rate, q. This tabular calculation will be made for the minimum expected air bleed temperature $T_4 = 80°F = 540°R$ that would be expected to leave the ECS heat exchanger. The minimum temperature, T_4, will result in the maximum expected air bleed flowrate, ω. The results are tabulated in the second column of Table 3-20.

From Table 3-19 it can be seen that the maximum by-pass fan bleed rate at sea level condition is $\omega_{max} = 2.3$ lb/sec. From the tabular solution in Table 3-20 it appears that one jet engine could supply the bleed air rates of $\omega \leq 1.286$ lb/sec required for $q \leq 80,000$ Btu/hr of ECS coolant loop heat rejection.

By-Pass Fan Bleed (10,000 ft) - High Efficiencies. For this condition, $p_1 = 27.9$ psia as indicated in Table 3-18 and $p_3 = 10.1$ psia, the atmospheric pressure at 10,000 ft altitude. Using only the root greater than one, the temperature ratio is:

$$\frac{T_3}{T} = 1.000 - 0.243 = 0.757$$

Assuming that the primary and secondary heat exchanger outlet temperatures are $T = 600°R$ even at this altitude, then T_3 would be:

$T_3 = -6°F \leq 15°F$. This is slightly less than T_3 at sea level for by-pass fan bleed, so that a slightly smaller air bleed flowrate, ω, would be required for this case as compared to the sea level tabular solution.

Since the ram air temperature at 10,000 ft altitude might be as high as $T_A = 60°F$, an outlet temperature from the heat exchangers would be about $T = 80°F = 540°R$. The turbine outlet temperature T_3 would then be: $T_3 = -51°F \leq 15°F$. This lower temperature T_3 expected at 10,000 ft altitude will nearly reduce the air bleed flowrates in half. Hence, the sea level and hot air condition would appear to require the maximum bleed air flowrate, ω.

Compressor Interstage Bleed: (Sea Level) - High Efficiencies. For this condition, $p_1 = 135$ psia.

$$\frac{p_3}{p_1} = \frac{14.7 \text{ psia}}{135 \text{ psia}} = 0.1089, \quad \left(\frac{p_3}{p_1}\right)^{\frac{k-1}{k}} = (0.1089)^{0.286} = 0.531$$

The turbine outlet temperature ratio is:

$$\frac{T_3}{T} = 0.600$$

C-3.

TABLE 3-20

JET ENGINE BLEED AIR FLOW RATES

HIGH EFFICIENCIES: $\eta_c = 0.60$, $\eta_t = 0.70$

COOLING RATE	FLOW RATE LB/SEC					
BTU/HR	BY-PASS FAN BLEED		COMPRESSOR INTERSTAGE BLEED		HIGH PRESS. COMP. BLEED	
	S. L.	10 K FT.	S. L.	10 K FT	S. L.	10 K. FT
20K	.32		.13		.10	.09
40K	.65		.26		.20	.18
60K	.97		.39		.29	.27
80K	1.29		.51		.39	.36
LOW EFFICIENCIES: $\eta_c = 0.40$, $\eta_t = 0.50$						
20K			.24		.17	.14
40K			.48		.34	.27
60K			.72		.50	.41
80K			.96		.67	.55

For a hot day T = 600°R at sea level, the T_3 would be:

$$T_3 = -100°F \leq +15°F$$

The air bleed flowrate ω required as a function of the ECS coolant loop heat rejection rate is shown in the third column of Table 3-20 based on an assumed temperature T_4 = 80°F as a minimum bleed air outlet temperature from the ECS heat exchanger.

The air bleed flowrates are $\omega \leq 0.514$ lb/sec, for $q \leq 80{,}000$ Btu/hr. From Table 3-19 this is much less the ω_{max} = 2.5 lb/sec air bleed for compressor interstage bleed from one jet engine. The air bleed rates at 10,000 ft altitude would be less than those shown above, to provide the same rate of refrigeration.

High-Pressure Compressor Bleed, Sea Level Hot Day, High Efficiencies.

$$p_1 = 418 \text{ psia}$$
$$\eta_c\, \eta_T = (0.60)\,(0.70) = 0.420$$
$$P_2 = 1038 \text{ psia}$$
$$\frac{T_3}{T} = .507$$

For the assumed T = 600°R at sea level - hot day conditions.
T_3 = -156°F ≤ +15°F
Assuming T_4 = 80°F, the flowrates in the third column result.

High Pressure Compressor Bleed, 10,000 ft Hot Day, High Efficiencies. According to Table 3-19, the ram air temperature, T_A, for a hot day at 10,000 ft altitude is T_A = 64°F. Let us assume an air bleed outlet temperature T = 100°F from the primary and secondary heat exchangers for this condition. This will correspond to the hot day, sea level conditions we have assumed in this study of $T_A \approx 105°F$ and T = 140°F, where $(T-T_A) \approx 35°F$. These values are probably greater than would actually occur from these heat exchangers, but this will result in a slightly higher computed value of ω, and hence a conservative solution. The values are shown in the second part of column 3 of Table 3-20

Calculations for Lower Efficiencies. The low bleed air flowrates, ω, computed for this system might result in low compressor and turbine efficiencies. The following calculations show the influence of lower values of η_c and η_r on the bleed air flowrate required for the by-pass fan bleed under hot, sea level conditions. Following the solution as before, assume $\eta_c = 0.40$ and $\eta_r = 0.50$ then

$$\frac{T_3}{T} = 0.869 \quad \text{and}$$

$$T_3 = +61°F \geq +15°F$$

Hence, with these low compressor and turbine efficiencies, $T_3 = 61°F$ is the coldest air temperature that could be supplied to the ECS heat exchanger, and Freon coolant temperatures could not be maintained below this temperature. About 4 times the air bleed flowrates shown in the table for the high efficiency sea level by-pass fan case would be required to reject the same q for this lower efficiency case. Hence, η_c and η_T values of at least 0.6 to 0.7 would appear necessary for this system.

The low compressor/turbine efficiencies would probably be acceptable when used with compressor interstage bleed under sea level, hot day conditions. The flowrates are shown in the second column of Table 3-20.

This low efficiency system results in about two times the air bleed flowrates as the higher efficiencies.

Hence, the low turbine/compressor efficiencies considered above could be used for air cycle refrigeration systems using either the compressor interstage bleed or the high-pressure compressor bleed. Only the by-pass fan bleed system would require turbine/compressor efficiencies of at least 0.6 to 0.7 for sufficient cooling under hot day, sea level condition.

The flowrates for the high pressure compressor bleed for sea level and 10,000-ft for hot day conditions and low turbine and compressor efficiencies are shown in the last columnes of Table 3-20.

These values indicate that the jet engines contemplated for the Shuttle have the capability to provide sufficient pressurized air flow for ECS cooling. This means that if this mode of cooling is selected for the ferry flights, probably two engines would be fitted with the appropriate mechanisms.

3.3.6.7.3 Location of Air Cycle Machine Coolant Loops. It would appear that the most logical location for the air cycle unit would be in the ferry engine nacelle. The cold air would have to be ducted about 100 feet (maximum) at low pressure to the EC/LSS coolant loop-air heat exchanger. Little or no insulation should be required on this line to obtain 40°F coolant loop temperatures, but potential frost and freezing problems exist in this line. The air supply line and the coolant loop-air heat exchanger would be aboard the Space Shuttle, all other parts of the system would be in a module on the ferry engine package.

Another possible design of the air cycle unit would be to mount the air cycle machine in the Space Shuttle cargo bay, or at the EC/LSS coolant loop location. A high-pressure, un-insulated air line would be required between the ferry engine for the bleed air. The air cycle unit would also require ram air cooling at its location if it were a turbine/compressor machine, whereas a turbine/generator machine would probably not require cooling at its location. The major disadvantage of this design would be the need to mount and dismount the air cycle machine separately from the ferry engines, and the probable need to supply ram air cooling ducts somewhere in the region of the crew cabin. Based upon the apparent disadvantages of this second possible system, it was decided to examine the system with the air cycle machine, as a unit module of the ferry engine package.

The air cycle unit would provide EC/LSS coolant loop heat rejection as long as the ferry engines were at least idling. To provide cooling when these engines are shut down, a heat exchanger would be required on the EC/LSS loop for GSE cooling. This GSE heat exchanger would be the same one used on the orbital mission launch pad, and a GSE coolant cost would have to be attached to this exchanger when the ferry flight engines were off at airports.

For the air cycle module located on the ferry engines package, two methods are possible for cooling the EC/LSS loop. The one already mentioned would be to run the cold air from the ferry engine module up to the EC/LSS coolant loop-air heat exchanger near the crew cabin. This would require a low pressure air line of rather large diameter, about 100 feet long, that would probably have to be insulated to some extent. The air temperature entering this line should not be much lower than 32°F because of ice formation problems at the turbine outlet. The air temperature leaving this duct must be < 40°F, in order to obtain 40°F coolant temperatures in the air-coolant heat exchanger. Hence, the thermal design of this cold air duct will be very restrictive, and considerable insulation weight might be required.

A second method of cooling the EC/LSS coolant loop would be to run the Freon 21 coolant lines back to the area where the ferry engine packages are attached to the Space Shuttle. The coolant loop-air heat exchanger would be located where cold air from the air cycle unit of the ferry engine package would blow through this heat exchanger. In this method, two coolant lines, small in diameter and insulated, would replace the cold air duct running between the EC/LSS and the ferry engines. These small diameter lines should be lightweight compared to a large diameter air duct, but the added Freon 21 liquid inventory in these long lines may offset any hardware weight advantage. These long lines could be valved off at the EC/LSS for mission phases that do not require air cycle cooling.

A variation of this cooling method would be to place the coolant loop-air heat exchanger on the ferry engine package. This system would remove the weight of this heat exchanger from the Space Shuttle for orbital missions. The major problem with this approach is that the EC/LSS coolant loop would have to be broken each time the ferry engine packages were installed or removed from the Space Shuttle.

3.3.6.8 Discussion of Heat Rejection System. Methods of providing heat rejection from the active coolant loops of the Space Shuttle during all expected mission phases have been briefly examined in the previous sections. The coolant

loops considered in those studies were the EC/LSS and the APU. The object of this section is to discuss the heat rejection systems that can be used with these coolant loops, and to point out some of the problem areas of these systems. The weight of these systems will be estimated in Section 3.3.6.9.

The EC/LSS coolant loop is expected to provide cooling for the manned cabin, life support systems, avionics compartment, fuel cells, and cabin windows. A maximum heat rejection rate of about 60,000 Bru/hr is expected to occur to this coolant loop. This heat rate will primarily be rejected from this loop by a radiator system on orbit. Additional heat rejection methods will be required for this coolant loop during the stowed radiator flight phases. The possible additional heat rejection systems for the EC/LSS coolant loop will be evaluated in this section. These heat rejection systems all make use of dedicated liquid evaporants, plus the possible use of ram air cooling in the lower atmosphere. These heat rejection systems must provide temperatures as low as 35°F for the EC/LSS coolant loop operation.

The APU coolant loop will provide cooling for the hydraulic oil, lubricating oil, electrical alternator, and turbine shields of the Auxiliary Power Units (APU). A total heat rate as high as 300,000 Btu/hr could occur to this coolant loop. Since the APUs will run during the ascent and de-orbit phases of the orbital mission, heat rejection systems using dedicated liquid evaporants, and possibly ram air cooling, will be considered. These heat rejection systems will be expected to maintain the APU coolant loop temperatures of about 150°F or lower during APU operation.

The EC/LSS and APUs could possibly use an integrated coolant loop, or separate coolant loops with an integrated heat rejection system, but these options will not be considered here. The EC/LSS and APUs have considerably different heat rejection requirements, so they will be considered separately. The EC/LSS will reject heat during all flight phases, with a low coolant loop temperature of about 35°F required and moderate heat rejection. The APUs will operate only during the ascent and de-orbit phases of orbital flight, with higher

heat rejection rates and coolant loop temperatures of about 150°F required. These differences in the coolant loop temperatures and heat rejection requirements are the main reasons for considering the EC/LSS and APU coolant loops, and possible heat rejection systems, separately.

The heat rejection systems that will be proposed for both the EC/LSS and APU coolant loops will be able to operate over all the proposed Space Shuttle flight phases. Any one of these heat rejection systems will be more optimum for certain flight phases, but they can all be used during orbital flight, ascent and reentry, aircraft flight, or ground hold conditions, if necessary. The most commonly considered heat rejection systems that could be used with (a) the EC/LSS coolant loop and (b) the APU coolant loop on the proposed space shuttle are:

(a) EC/LSS Coolant Loop:

(1) Hydrogen Heating and Venting
(2) NH_3 + Water Evaporation
(3) Water Evaporation/Ram Air Cooling of Vapor Compression Refrigeration Cycle

(b) APU Coolant Loop:

(1) Hydrogen Heating and Venting
(2) NH_3 + Water Evaporation
(3) Water Evaporation/Ram Air Cooling

The three heat rejection systems appear to be the same for both the EC/LSS and APU coolant loops. The third system, the combined water evaporation/ram air cooling, would require an active vapor compression refrigeration cycle to obtain the low temperatures needed for the EC/LSS coolant loop. Considerations relating to these three heat rejection systems are discussed below. They are discussed in some detail; however, weight estimates were for slightly different arrangements, as discussed in the next section.

3.3.6.8.1 Hydrogen Heating and Venting. This system rejects heat to expendable hydrogen gas, which is stored at cryogenic temperatures as either subcritical LH_2 or supercritical LH_2. This system makes use of the large enthalpy change available when hydrogen is warmed from cryogenic temperature to a warm vented gas, where the enthalpy change ranges from 1500 to 2100 Btu/lb. Hence, the mass of dedicated liquid that must be vented is a minimum for this sytem compared to others. The hydrogen/coolant heat exchangers need only be designed for GH_2 flow, and they should be small and lightweight because of the large ΔT heat transfer processes.

This hydrogen warming and venting system also has some problem areas. The most important is the hazardous nature of hydrogen, especially when vented into the earth's atmosphere. Although it is non-toxic and non-corrosive, explosive mixtures of hydrogen and air can exist over a range of wide mixture ratios. The hydrogen vent system must be designed carefully to eliminate combustible mixtures or to burn the hydrogen gas in the atmosphere. The system must also be designed to eliminate leakage or LH_2 spills during filling.

Another potential problem is the logistics of LH_2 supply, especially for the early horizontal tests and ferry flights, with an LH_2 supply required at each airport. The low density of LH_2 is a slight disadvantage, in that larger weight tanks will be required to contain the LH_2, as compared to more conventional liquids.

Thermal stress problems must also be considered in the design of the hydrogen heat exchangers. With cryogenic hydrogen inlet temperatures, large ΔTs within the heat exchanger, and transient operation, this heat exchanger will be prone to thermal stress problems. Although little experience exists in the design of warm fluid/cryogenic heat exchangers, the stress and transient problems in conventional heat exchangers are fairly well understood. The problems of potential freezing and control of the warm coolant outlet temperature from a cryogenic heat exchanger, are areas that will require consideration in the design of this heat rejection system.

A heat rejection system that would make the best of use of expandable liquids on the Space Shuttle would be a combined hydrogen warming and venting plus water evaporation system. This system would utilize possible expendables such as water and fuel cell LH_2, so that a minimum weight of dedicated liquid evaporants would have to be loaded for a mission. The high latent heat of vaporization of water, its non-toxic, non-flammable and non-corrosive nature, and its general avilability, make water a very desirable expendable evaporant.

However, the low vapor pressure of water at the desired coolant loop temperature does not allow it to evaporate or boil in the lower atmosphere or at sea level. Hence, a combined H_2 plus H_2O venting heat rejection system would have to vent hydrogen in the lower atmosphere, whereas water could only be evaporated in the upper atmosphere, or in space. Compared to the H_2 warming venting system, the combined H_2 plus H_2O venting system would require a water boiler that would operate from zero-g to a few g's acceleration. Hence, the fixed weight of the combined H_2 plus H_2O venting system would be greater than that of the H_2 venting system, but less dedicated liquid would be required. The combined system could utilize fuel cell H_2O as expendables.

3.3.6.8.2 <u>Ammonia Plus Water Evaporation.</u> This sytem would reject heat to the expendable liquid evaporants, water and ammonia. Water is a desirable liquid evaporant because of its high latent heat, availability, safety, and other desirable properties. A supply of expendable liquid water is available on the Space Shuttle, and this could be utilized in this heat rejection system. However, the low vapor pressure of water at the desired coolant loop temperature does not allow it to evaporate in the lower atmosphere. Hence, a second liquid evaporant is required to reject heat in the lower atmosphere or at sea level conditions.

Ammonia appears to be the best expendable liquid evaporant that could be used for heat rejection in the lower atmosphere. Its latent heat of vaporization is moderatly high (500 Btu/lb) but compared to water, about twice the mass of ammonia must be evaporated to obtain the same amount of heat rejection. This system would require zero to a few g's acceleration boilers for both the liquid

water and ammonia evaporation. To minimize the weight of liquid evaporants and the weight of dedicated liquids required, this heat rejection system should be utilized so that a maximum amount of water is evaporated for any mission profile of heat rejection.

The properties of ammonia will create problems in its use as an expendable liquid evaporant. Ammonia vapor is toxic to humans, even in small concentrations in the atmosphere. It is also corrosive to certain metals, especially to copper and its alloys. With careful design of vents, it should be possible to safely vent ammonia into the Space Shuttle environment, even with humans present in an airport environment.

The logistics problem of ammonia supply will also be present, especially during the early horizontal test and ferry flights. Ammonia can only be used for heat rejection in the lower atmosphere, so that each Space Shuttle airport must have liquid ammonia available. For horizontal test and ferry flights, the dedicated ammonia weights will be heavy.

Control of the water and ammonia boilers in this heat rejection system should also be considered. For mission planning, it would always appear advantageous to maximize the water evaporation. During a mission, however, the evaporants should be used to maximize the potential cooling available during the remainder of the mission. The outlet temperature of the coolant flowing through these evaporators will be controlled by regulating the vapor pressure and rate of evaporation of the expendable liquid evaporants in each of the liquid boilers.

3.3.6.8.3 Water Evaporation/Ram Air Cooling. This system will reject heat to evaporating water in space or the upper atmosphere, whereas heat rejection will occur to a ram air coolant stream in the lower atmosphere or on the ground. The major advantage of this heat rejection system is that it uses the two most common fluids as coolants, water and air. Hence, this system would have no toxicity or explosion problems, and only minor corrosion or venting problems could exist with this system.

This system will reject heat directly from the APU coolant loop at temperature $T \leq 150°F$, with the water evaporation to ram air change in cooling occurring at an altitude of about 35,000 ft. For the EC/LSS coolant loop, with heat rejection in the altitude range 35,000 ft to 150,000 ft. To obtain continuous heat rejection for the EC/LSS loop, it will be necessary to raise the heat rejection temperature to 150°F. This can be done by placing a vapor compression refrigeration cycle between the EC/LSS coolant loop and the combined water evaporation/ram air cooling heat rejection system. The vapor compression refrigeration cycle would add weight and power penalties to the Space Shuttle, but this would only be required for the EC/LSS coolant loop.

A water boiler that could operate in the acceleration range of zero to a few g's would be required for the water evaporator heat rejection. This boiler would be located directly on the APU coolant loop. For the EC/LSS coolant loop with vapor compression refrigeration cycle, a second water boiler would be required at the condenser to provide cooling in the altitude range of 35,000 ft $< h <$ 150,000 ft. This boiler would operate under near standard gravity conditions, and might be as simple as a water spray on the refrigeration cycle condenser. Control of the water evaporation pressure and water supply to these boilers will control the heat rejection temperature and heat rejection rate in these water evaporators.

For altitudes below 35,000 ft, heat rejection in this system would occur to a ram air cooler. For the APU coolant loop, this ram air cooler would be placed in series with the water boiler on the coolant loop. For the EC/LSS coolant loop, the ram air cooler would be located at the refrigeration cycle condenser. In either case, the ram air coolers would be of the finned tube design, commonly used in automobile radiators and aircraft oil coolers. Ducting with fans during ground-hold would be required to supply the ram air to the coolers. The heat rejection rate and temperature of these coolers would be controlled by the ram air coolant flowrate.

To reduce the weight of this heat rejection system, it might be possible to build a combined water evaporation/ram air cooler for either the coolant

loops or the refrigeration cycle condenser. This would be a finned tube heat exchanger, where either ram air or a water spray would cool the fin surfaces. A zero-g water spray, vapor swirl evaporator design might limit water loss to an acceptable level. If this is not possible, the technology exists to make separate water evaporation and ram air coolers for this system.

The early horizontal test and normal ferry flights will occur at altitudes h < 20,000 ft. Hence, this heat rejection system would use ram air cooling for these long-term flight phases, and dedicated liquid evaporants would not.

3.3.6.9 Weight Estimates of Possible Heat Rejection Systems. Weight estimates of the most reasonable heat rejection systems are given in this section. The systems selected were based on the discussions in the previous section.

The first one, the hydrogen venting system, will undoubtedly be the lightest weight of the three. The second system, ammonia plus water evaporation, would be much heavier and would also vent a hazardous fluid. The third system, which rejects heat by water evaporation/ram air cooling, utilizes inert fluids which would reduce safety problems. This system may also provide weight advantages for the horizontal test and ferry flight phases of the Space Shuttle mission. Based upon these considerations, as well as those in the previous section, it was decided to make a more detailed comparison of the hydrogen venting and the water evaporation/ram air cooling systems. The object of this section is to make weight estimates of the three heat rejection systems listed below:

1. Hydrogen Heating and Venting for EC/LSS and APU
2. Water Evaporation/Ram Air Cooling for APU
3. Water Evaporation/Ram Air Cooling with Refrigeration Cycle for EC/LSS

These three systems can be combined in various ways to provide EC/LSS and APU cooling.

The scar weight associated with a jet engine bleed air cycle machine for ferry flights is also given.

The above three heat rejection systems are shown schematically in Figs. 3-44, 3-45 and 3-46. These systems employ the basic elements required to obtain the necessary heat rejection, but they do not include any back-up means of heat rejection. As shown in the figures, these systems have no heat rejection redundancy, and as such offer a fail-hot operational capability. A second parallel set of these heat rejection elements would have to be included for any of these three systems to obtain a fail-operate/fail-hot redundancy capability. Three parallel heat rejection elements would be required to obtain fail-operate/fail-operate redundance capability, but it is doubtful that the coolant loops will require this degree of heat rejection redundancy.

The APU coolant loops, with separate loops for each APU, should only require a fail-hot capability. The APU heat rejection system, with some degree of integration of the coolant loops and/or heat rejection elements, would probably require a fail-operate/fail-hot capability. A completely integrated APU coolant system would require fail-operate/fail-operate capability, and this degree of redundancy for a completely integrated APU coolant system would appear unrealistic.

The EC/LSS coolant loop will at least require a heat rejection capability of fail-operate/fail-hot. If a life-threatening environment were to exist in the crew cabin due to a failure of the EC/LSS heat rejection system, then a fail-operate/fail-operate redundancy capability would undoubtedly be required for the EC/LSS coolant loop. Hence, the EC/LSS coolant loop will require at least two, or possibly three, parallel heat rejection elements to obtain the necessary operational safety.

For the purposes of weight estimates it will be assumed that all of these heat rejection systems will be designed for a fail-operate/fail-hot capability, so that two parallel heat rejection elements will be used in each of the systems. One supply of expendable liquid evaporant and its associated tankage will be assumed to be shared by the two heat rejection elements. Let us first estimate the weight of the heat rejection elements and the total system hardware, and then we can estimate the weight of the expendable liquid evaporants and the weight penalty of the associated tankage.

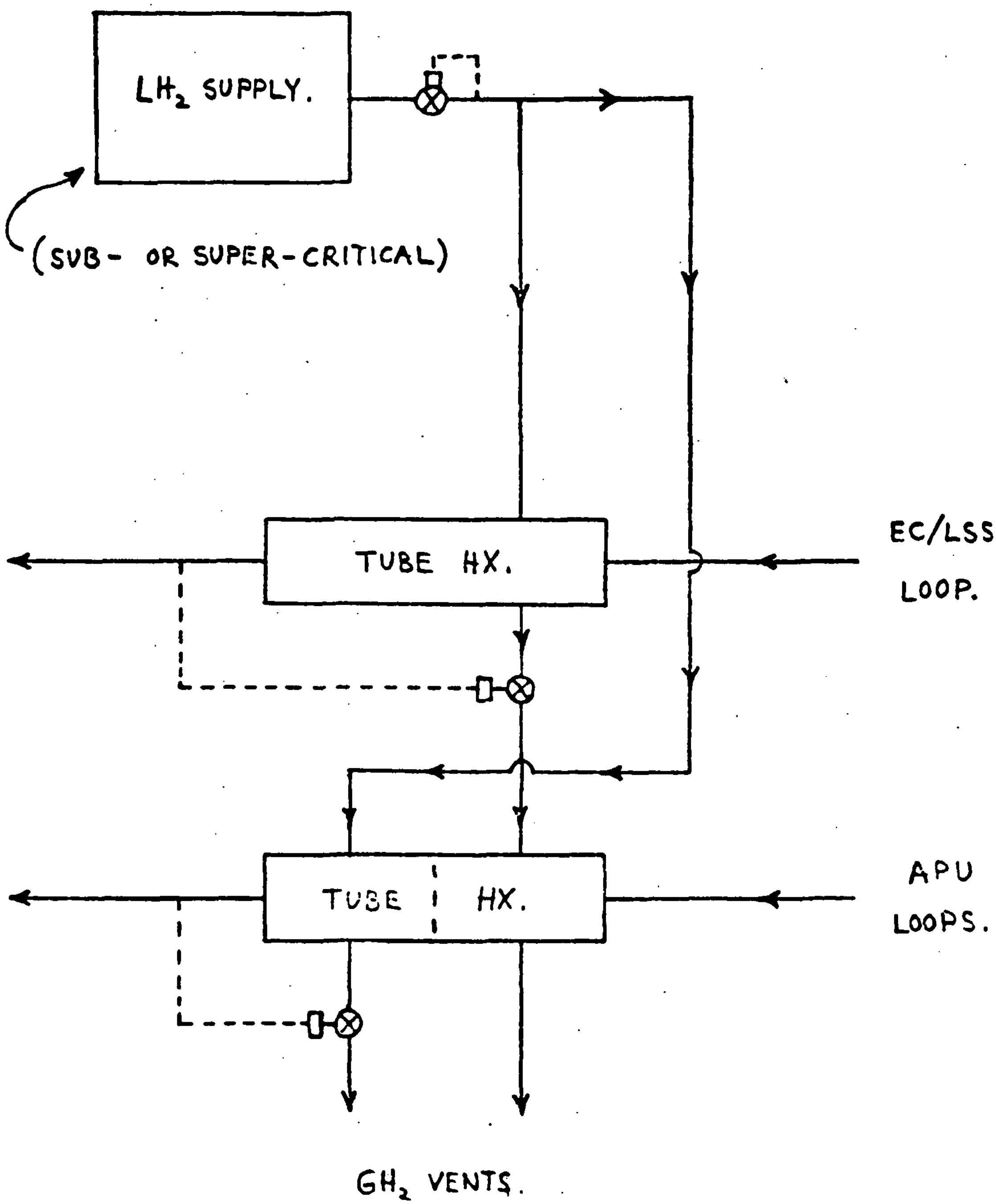


Fig. 3-44 Hydrogen Heating and Venting System for EC/LSS and APU

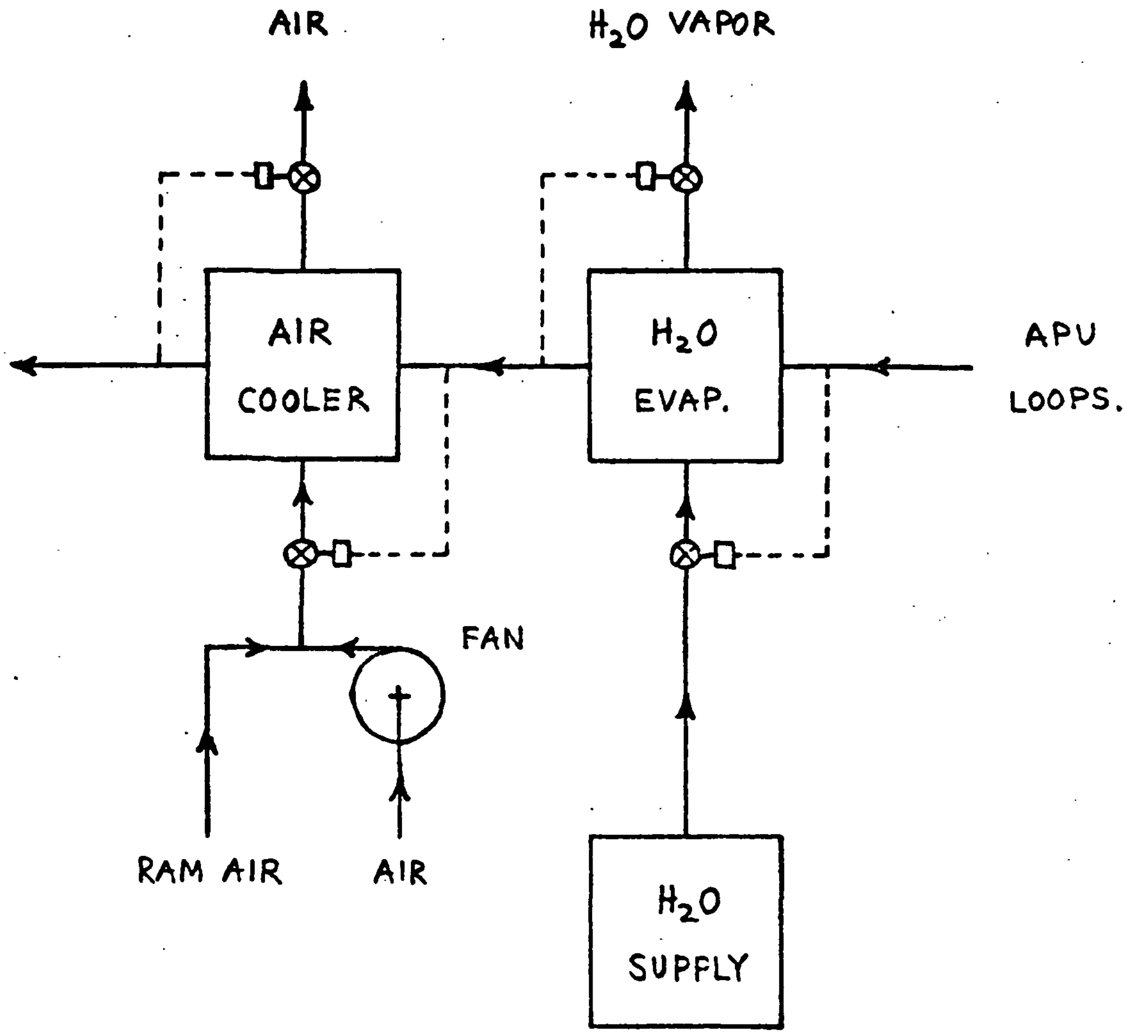


Fig. 3-45 Water Evaporation/RAM Air Cooling System for APU

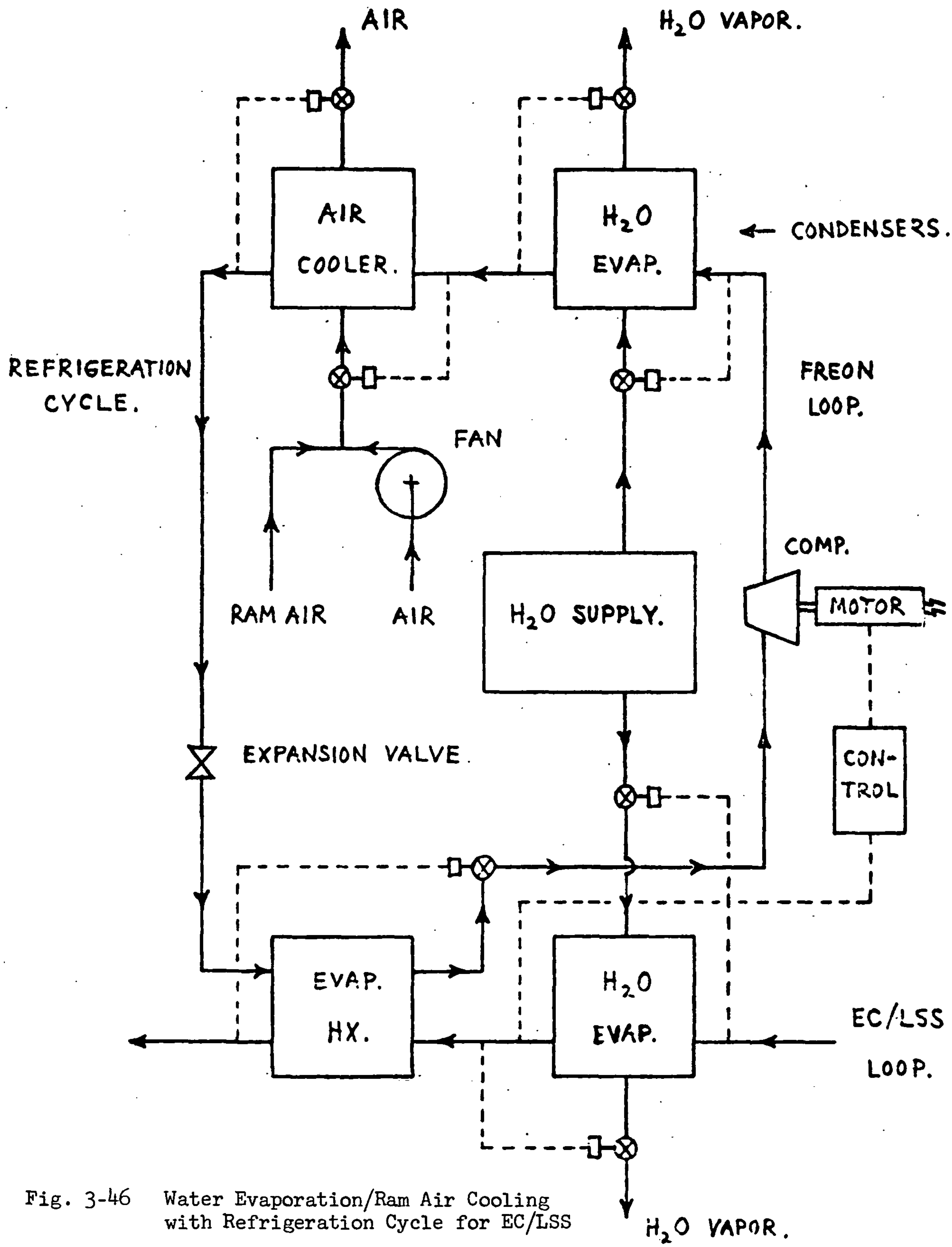


Fig. 3-46 Water Evaporation/Ram Air Cooling with Refrigeration Cycle for EC/LSS

3.3.6.9.1 <u>Hydrogen Heating and Venting System for EC/LSS and APU.</u> One heat rejection element for this system is shown in Fig. 3-44. The GH_2 coolant loop heat exchangers are assumed to be of the lightweight concentric tube design, with a maximum design heat rate of 90,000 Btu/hr for the EC/LSS heat exchanger and 280,000 Btu/hr for the APU heat exchanger. The APU heat exchanger is also assumed to be a GH_2 high-pressure oil heat exchanger. A weight estimate of one heat rejection element is as follows:

EC/LSS Heat Exchanger and Controls	10 lb.
APU Heat Exchanger and Controls (High pressure oil)	40 lb
Tubing, Insulation, GH_2 Vents	10 lb
Total Element Hardware	60 lb

The total system hardware weight of two heat transfer elements would be

Total System Hardware:	120 lb

Only small amounts of electrical power would be required to operate this heat rejection system; hence no weight penalty should be involved for an electrical power source.

The required LH_2 loadings and the weights of two possible LH_2 storage tank systems will now be estimated for this system, followed by an estimate of the total system lift-off weight.

The LH_2 expendables required for the orbital mission are 170 lb for APU cooling (400,000 Btu heat rejection) and 118 lb for EC/LSS cooling (200,000 Btu). These values take into account system capacitances and the extra heat capacity of the hydrogen from the EC/LSS to the APU. For the ferry mission, with time period of 10 hrs maximum, and an average heat rate of 45,000 Btu/hr required for the EC/LSS, a LH_2 loading of 265 lb would be required. Hence, the maximum expected LH_2 loadings for this system would be:

LH_2 Mass (Orbital Mission) = 170 lb + 118 lb = 288 lb

LH_2 Mass (Ferry Flight) = 265 lb

There is sufficient hydrogen capacity, as required by the APU and EC/LSS orbital operations, to contain the hydrogen for the EC/LSS heat rejection during the ferry flights. The tank system weight for the required 288 lb of usable LH_2 would be 420-lb, based on a 10 percent reserve and a 40-lb residual. This is for supercritical tankage and includes insulation, vacuum jackets, supports, etc. For a subcritical LH_2 tankage system, with vacuum jacketed insulation systems, a tankage system weight of about 300 lb would result.

For the orbital mission, the hydrogen heating and venting heat rejection system for both the EC/LSS and APU coolant loops would have a lift-off weight in the range 800 to 900 lb. For a short-duration, 2-hour ferry flight, the total system weight at lift-off should range from 500 to 700 lb, depending on the type of LH_2 storage system utilized for the hydrogen heat rejection system.

3.3.6.9.2 Water Evaporation/Ram Air Cooling for APU. A heat rejection element for this system is shown in Fig. 3-45. A maximum APU heat rejection rate of 280,000 Btu/hr is assumed for system design. An oil/air finned tube heat exchanger is assumed for the ram air cooler. A zero-gravity water boiler would be needed for the water evaporator, and the weight estimate shown below for this rather heavy unit is only a rough estimate. The ram air ducting and fan would probably also have considerable weight, depending upon its design and location in the Space Shuttle. A weight estimate of one heat rejection element is given below:

Oil/Air Heat Exchanger and Controls	30 lb
Ram Air Ducting, Controls and Fan	50 lb
Water Evaporator and Controls	60 lb
Tubing, H_2O Vapor Vents, etc.	10 lb
Total Element Hardware	150 lb

The total system hardware weight with two heat transfer elements would be 300 lb.

The ram air fan would probably require considerable electrical power during ground hold operation. An upper estimate would be about 8 kW.

The water evaporation/ram air cooling system for the APU will have to reject 360,000 Btu by water evaporation, which will require about 360 lb of water, for the orbital mission. For the ferry flights the APUs would not be used and no water would have to be loaded.

The water tankage for this APU heat rejection system should be designed to store 360 lb of usable water plus 10 percent for reserve and 10 lb residuals for a total of 406 lb. Aluminum water tanks with about 0.025-inch walls, tank supports, bladders, pressurization, etc., would weigh about 50 lb.

The total lift-off weight is 756 lb, the sum of the system hardware, tankage, and expendable liquid water.

3.3.6.9.3 Water Evaporation/Ram Air Cooling with Refrigeration Cycle for EC/LSS. Figure 3-46 shows a heat rejection element for this system. A maximum heat rejection rate of 90,000 Btu/hr was assumed for the EC/LSS coolant loop and 135,000 Btu/hr from the refrigeration cycle. The weight estimate shown below for this refrigeration cycle heat rejection element corresponds well with known weights of aircraft vapor compression refrigeration cycles.

Air Cooled Condenser and Controls	15 lb
Ram Air Ducting, Control and Fan	25 lb
Water Evaporation Condenser and Controls	30 lb
Water Evaporation Cooler and Controls	30 lb
Water Tubing and H_2O Vapor Vents	10 lb
Evaporator Heat Exchanger	15 lb
Compressor and Electric Motor	100 lb
Freon Mass, Tubing, Valves, etc.	25 lb
Total Element Hardware	250 lb

The total system hardware weight for two heat rejection elements would be 500 lb.

A maximum total estimate of electrical power for the compressor motor (18 hp) and the ram air fan (5 hp) would be about 17 kW.

Heat rejection by water evaporation should total 170,000 Btu for the EC/LSS coolant loop using the water evaporation/ram air cooling system with refrigeration cycle. Hence, 170 lb of expendable liquid water with 10 percent reserve and 5 lb residuals, for a total of 192 lb, would be required for the orbital mission. For the ferry or other atmospheric flights, only ram air cooling would be utilized, and no liquid water would have to be loaded. A tankage weight of 30 lb is estimated to contain this water.

The total lift-off weight of this system would then be 722 lb. The weight penalties associated with the electrical power requirement of this system are not included, although the power requirement of about kW for the refrigeration cycle compressor and ram air fan is considerable.

For atmospheric flights with ram air heat rejection, a lift-off weight of about 530 lb should exist for this EC/LSS water evaporation/ram air cooling system with a vapor compression refrigeration cycle.

3.3.6.9.4 Weight Comparison for Both EC/LSS and APU Heat Rejection Systems. The first hydrogen heating and venting system shown in Fig. 3-44 provides heat rejection for both the EC/LSS and APU coolant loops. The water evaporation/ram air cooling systems sketched in Fig. 3-45 and 3-46, provide heat rejection for the APU and EC/LSS coolant loops, respectively. A water evaporation/ram air cooling system for both the EC/LSS and APU should have a weight estimate near the sum of the two independent systems shown on Fig. 3-45 and 3-46. Some weight would probably be saved by integration of the water supplies, ram air coolers, and possibly some water evaporators for a combined heat rejection system. However, by adding their weights for a combined EC/LSS and APU water evaporation/ram air heat rejection system, a conservative weight estimate for a total water evaporation/ram air cooling system would result.

A third possibility for a total EC/LSS and APU heat rejection system, which would appear to offer a low weight possibility, would be hydrogen cooling of the EC/LSS coolant loop and water evaporation/ram air cooling of the APU coolant loops. The hydrogen EC/LSS coolant loop would be similar to that of Fig. 3-44, except the APU heat exchanger would not be required. The LH_2 storage tanks required would also be smaller, so a weight estimate of an EC/LSS hydrogen heat rejection system would be as follows:

	Supercritical Storage	Subcritical Storage
Total System Hardware	50 lb	50 lb
LH_2 Mass (Max.)	180 lb	139 lb
LH_2 Tankage	420 lb	300 lb
Total Lift-off Weight (Max.)	650 lb	489 lb

The above maximum LH_2 loading is that required for EC/LSS cooling during orbital flight and the tanks are sized for a 10-hour ferry flight. The weight estimate for the APU water evaporation/ram air cooler for this system would be the same as those already determined for System 2 (Fig. 3-45).

Three total heat rejection systems will be considered here for both EC/LSS and APU heat rejection, as below:

A. Hydrogen Heating and Venting for EC/LSS and APU.

B. Water Evaporation/Ram Air Cooling for EC/LSS and APU.

C. Hydrogen Cooling for EC/LSS and Water Evaporation/Ram Cooling for APU.

Weight estimates for the total system (A above) can be made from the data for the all hydrogen system. Weight estimates for the second system (B above) are totalled from the APU and EC/LSS system weights derived

separately. Weight estimates for the third system (C above) can be made from the EC/LSS system estimates and the APU cooling system weights.

The lift-off weights for these three total heat rejection systems were computed for three mission phases — the orbital mission, the 10-hour ferry flight, and the 2.5 hour ferry flight. The total lift-off system weights for these three heat rejection systems and three mission phases are shown in Table 3-21.

The results of the above weight table (Table 3-21) show that for the orbital mission, the hydrogen heat rejection, System A, is considerably lighter than the other two. For the long-term, 10-hour, ferry flight, the water evaporation/ram air cooling, System B, appears to be competitive with System A. For the short-term horizontal test or ferry flights of about 2 hours duration, the hydrogen System A appears to offer a weight advantage over Systems B and C. The hybrid System C appears to result in lift-off weights comparable to those of Systems B, except for the long-term ferry flight where its lift-off weight is heavier. There would appear to be little reason to further consider System C, except that the development of the APU cooling system could proceed independently from that of the EC/LSS.

The hydrogen System A is, of course, lighter than System B for all flight phases. For the important orbital mission, System A is about 500 lb lighter than the water evaporation/ram air cooling System B. However, for the ferry flight missions, the weights of the two systems are comparable.

Table 3-21

TOTAL HEAT REJECTION SYSTEMS LIFT-OFF WEIGHTS, LBS

SYSTEMS	System A		System B	System C
	LH_2			
MISSIONS	Subcritical	Supercritical	Water/ Ram Air	LH_2EC/LSS H_2O APU Air
Orbital Mission	777	897	1410	1406*
Ferry Flight (10 hr)	754	874	880	1152*
Ferry Flight (2.5 hr)	544	664	880	933*

*NOTE: The LH_2 tankage weights for EC/LSS hydrogen heat rejection were assumed to be the heavier supercritical variety for System C.

This weight advantage of System A is offset to some extent by the logistics and hazard problems associated with this hydrogen heat rejection system. The water evaporation/ram air System B does not have any logistics or hazard problems, as would be the case with any other liquid evaporant heat rejection system. Although a weight study was not made, System B should be lighter than a pure ammonia evaporation system. System B should be competitive weight-wise with a combination water/ammonia system and it will show distinct weight advantages over an ammonia evaporation system for the horizontal test and ferry flights. The ammonia and/or water/ammonia evaporation heat rejection systems also have hazard and logistics problems.

If the lightweight hydrogen cooling System A could not be used on the Space Shuttle because of hazard and/or logistics problems, then the water evaporation/ram air cooling System B should be developed for all flight phases. This system would appear to be as lightweight as any other alternate system, and it does not offer any hazard or logistics problems.

The possibility also exists to use a water evaporation/ram air system for EC/LSS cooling during the initial horizontal test and ferry flights in the atmosphere, and switch to a hydrogen or a hydrogen/water system for orbital missions. This combination would then be able to utilize the significant weight advantage that a hydrogen system shows for the orbital mission, but not imperil the Space Shuttle during the early atmospheric test or ferry flights.

3.3.6.9.5 Weights for Jet Engine Bleed/Air Cycle Cooling. The systems previously discussed are applicable to both orbital and ferry flights. Another option, as discussed in Section 3.3.6.7, is the use of a jet engine bleed air cycle machine that would be installed when the ferry engines are installed.

A weight penalty estimates of some of the most promising air cycle heat rejection systems are given below. It is assumed the EC/LSS coolant loop must have a fail-operate/fail-safe heat rejection capability. This degree of operational safety would no doubt require two EC/LSS coolant loops and two bleed air cycle machines. Most studies to date have indicated that system weights for atmospheric flight phases are not too important a consideration in the design of

Space Shuttle systems. However, the weights and penalties of systems and components that are used during the orbital mission phase are very important considerations. Hence, this weight analysis of the air cycle heat rejection systems will concentrate on the orbital weight penalty of the various systems on the Space Shuttle. The weights of hardware included on the ferry engine modules, and are only present for atmospheric flights, will be only roughly estimated.

The most promising air cycle heat rejection systems that were considered previously, would all have the air cycle machines located in the ferry engine package. Hence, these units would not be present for the orbital mission. It is assumed that one air cycle machine is mounted on each of the two ferry engines that would be mounted on the Space Shuttle for atmospheric flights. These air cycle machines would supply cold air for cooling of the EC/LSS coolant loop. A schematic diagram of one of these air cycle machines is shown in Fig. 3-43.

This air cycle machine, mounted in the ferry engine nacelle, should be able to use bypass fan air of the ferry engines to cool the compressor bleed air heat exchangers. Hence, cooling of the compressor bleed air heat exchangers will occur whenever the ferry engines are at least idling, and no ram air fans or ducts should be required for this application. The cold outlet air temperature is controlled by bypassing warm compressor bleed air around the compressor/turbine assembly.

A rough estimate of the weight of this air cycle machine can be obtained from AiResearch data in Report No. 77-7815. That air cycle unit afforded about 55,000 Btu/hr of cooling, with a total estimated weight of about 135 lb. The primary and secondary bleed air heat exchanger assembly will be rather heavy, because it will have to be made of steel to withstand the high compressor bleed air temperatures from the ferry engine. A compressor/turbine assembly, of about 5 inch rotor diameter, should be required for this air cycle. Weight estimates of these components plus the ducting and control valves associated with one air cycle machine are shown below:

Bleed Air Heat Exchange Assembly	60 lb
Compressor/Turbine Assembly	50 lb
Ducting and Valves, Controls, etc.	40 lb
Total Weight of One Air Cycle Machine	150 lb

The above weight estimate is for one air cycle machine mounted on one ferry engine module. Since each of the two ferry engine modules would have its own air cycle machine, the total weight penalty of the air cycle machines on the ferry engine modules for atmospheric flight would be 300 lb. This 300-lb weight penalty would be present only when the ferry engines are installed for horizontal test and ferry flight phases, and would not be present for the orbital mission. Let us now estimate the weight penalties that must be built into the Space Shuttle, and will be present for all missions, for various systems that would afford air cycle machine cooling of the EC/LSS coolant loops.

The first system would duct the cold air from the air cycle machines to the EC/LSS coolant loop near the crew cabin. A liquid coolant/air heat exchanger would be mounted on the EC/LSS loops to provide heat rejection to the air cycle air. This heat exchanger would be of the typical finned tube design common for aircraft, and it should contain both the primary and secondary coolant loops. The total weight of this heat exchanger would be about 25 lb, for a total heat rejection rate of 60,000 Btu/hr.

The cold air duct would run about 100 feet between the ferry engines and the crew cabin area. For a cold air bleed flowrate of from 1 to 2 lb/sec., a duct diameter of about 6 inches should be required to limit pressure drops. This duct would probably have to be insulated with about 1/2 in. of spray-on foam insulation. This system would be prone to ice formation in the cold air duct, and it might be necessary to have an ice separator at the air cycle machine air outlet. The weight of this duct and insulation would be about 60 lb.

The total weight of this system, which delivers cold air to an EC/LSS heat exchanger near the crew cabin, would be the sum of the weights of the air line, coolant/air heat exchanger and controls, and amounts to about 90 lb.

The weight of the air line on this system is rather heavy. Although this system would have two air cycle machines and two EC/LSS coolant loops built into the integral coolant/air heat exchanger, only one line would appear to be required. The air output from either air cycle machine could be run through this line using rather simple flap valves, with the air cycle outlet pressure forcing the flow. A second air line would appear to be too large a weight penalty. The liquid coolant outlet temperature from the coolant/air heat exchanger could be controlled by bypassing some of the cool air flow.

A second system for air cycle cooling of the EC/LSS loops would be to run the liquid coolant loops back to the location where the ferry engines are mounted on the Space Shuttle. The coolant/air heat exchanger would be located on the Space Shuttle, with cold air from the air cycle machines cooling this heat exchanger. Compared to the first system, this second system substitutes the liquid coolant lines for the cold air duct weight. For a Freon 21 coolant flowrate of about 1200 lb/hr, coolant lines of about 1/2 in. diameter should yield reasonable pressure drops over long length loops. Each coolant loop would require two lines, each about 100 ft long, to extend the loop back to the ferry engine mounts. The weight on one coolant loop extension, assuming 0.024-in. thick aluminum tubing with 1/2-in. thick foam insulation, would be about 40 lb.

If both EC/LSS coolant losses are extended back to the ferry engine mounts, the total lines weight would then be about 80 lb.

This system, with both coolant loops extended and with two independent coolant/air heat exchangers and controls located at the two ferry engine mounts, would weigh about 120 lb. This system would appear to be heavier than the first cold air duct method of EC/LSS cooling, which had a total weight estimate of 90 lb.

This last coolant loop extension system has two completely independent methods of cooling the EC/LSS. The first cold air duct system utilized the common cold air duct, so that system would not have the same degree of redundancy. To install two completely independent cold air ducts and EC/LSS cooling paths for the first system would require a weight penalty on the Space Shuttle of about 150 lb.

The air cycle cooling systems for the EC/LSS loops considered above have a weight penalty in the range of 90 to 150 lb for the Space Shuttle. A further weight saving could be made for the Space Shuttle if the liquid coolant/air heat exchangers could be mounted on the ferry engine modules. This system would require breaking the coolant lines each time the ferry engines were mounted or dismounted from the Space Shuttle, but the coolant/air heat exchangers and their controls would be removed from the Space Shuttle air frame. Quick disconnect couplers, with no leakage and virtually no spill or air entrainment are available for this application. These couplers weigh about 1.0 lb each, and they should allow the coolant lines to be broken and re-attached without purging the Freon 21 loops or checking their operation. If this could be done, the weight penalty on the Space Shuttle using two independent coolant loop extensions would reduce to 85 lb, assuming 5.0 lbs. for couplers and valves.

Hence, these systems that perform air cycle machine cooling of the EC/LSS loops during atmospheric flights would appear to place a weight penalty of from about 90 to 150 lb on the Space Shuttle body, which would also be carried for the orbital mission.

Based upon the previous weights, the hydrogen heating and venting heat rejection system had a maximum loaded takeoff weight penalty of from 800 to 900 lbs for EC/LSS and APU cooling for the orbital mission. If the air cycle heat rejection system for the EC/LSS considered in this section is added to the Space Shuttle for the horizontal test and ferry flights, the total lift-off heat rejection weight penalty for the orbital mission with LH_2 cooling would be about 1000 lb. This weight penalty is less than the

1400 lb computed for the orbital mission water evaporation/ram air cooling system. The lift-off weight penalty of a water/ammonia evaporation heat rejection system for the orbital mission would be considerably greater than 1000 lbs.

The ferry engine-mounted air cycle cooling systems considered above will allow the horizontal test and ferry flights to be performed without loading LH_2 on the Space Shuttle. The small weight penalty on the Space Shuttle for this air cycle system will not seriously degrade the weight advantage of the LH_2 heating and venting system for the orbital mission. The combination of these two heat rejection systems should result in the lowest orbital weight penalties and the most logical atmospheric flight system of any heat rejectors considered.

Section 4

CONCLUSIONS AND RECOMMENDATIONS

The original objective of this study was to find ways of eliminating the radiators by the use of on-board cryogens. This objective was not completely achieved because of the change in vehicle configuration. However, the preliminary work that was completed gave a strong indication that a weight advantage would not result by eliminating the radiators and instead transferring the heat to the ACPS and fuel cell reactants and to additional dedicated hydrogen. The primary reason for this is the incompatibility of the heat being generated and the use rate of the cryogens that would absorb this heat. Long periods of time exist when power is required but very little ACPS propellant is used. Thus, the fuel cells and the electronics would continue to generate heat but very little "heat sink" was being expended.

When it was attempted to eliminate this incompatibility by storing the heated reactants in accumulators, for use later, an even greater penalty resulted. The accumulators became very large in order to store enough of the heated gas to be useful.

Hence, it is concluded that this particular approach would not be beneficial and the radiators should not be eliminated.

Other studies indicated that cryogens can play a useful fole for supplemental cooling.

It appears that oxygen and hydrogen will be used for the propellants during the ascent phase and therefore a heat sink exists during this time. The propellants and residuals have the capacity to absorb all of the EC/LSS generated heat. The system to transfer the heat from the EC/LS system to the ascent tanks is relatively simple. It would consist of Freon 21/cryogen heat exchangers, circulators, and controls, and would be relatively light. If it becomes desirable to not vent expendable fluids that would be required by more conventional EC/LSS cooling systems during lift-off and

ascent, then the heat could be transferred to the ascent tanks. However, since the EC/LSS will require a cooling system for the descent phase of flight (that does not depend on the drop tanks) which can also be used for the ascent phase, then the slight weight savings of the expendable coolant does not warrant the additional complexity required to transfer the heat to the ascent tanks.

Studies by the AiResearch Manufacturing Company on Freon 21/cryogen heat exchangers, as they relate to the various Shuttle requirements, indicate that these heat exchangers can be built, are lightweight, compact, and should present few technology problems. It is suggested that development of these heat exchangers be continued.

Comparison studies between three different types of APUs and the related cooling requirements indicates that an APU which uses cryogenically-stored oxygen and hydrogen provides the lightest overall system. The cryogenically stored oxygen and hydrogen can be used to absorb the heat generated by the APU operation as well as the heat generated by the EC/LSS.

It is suggested that serious consideration be given to utilizing an oxygen/hydrogen APU instead of the hydrazine APU currently planned. At least the current oxygen/hydrogen APU technology programs should be continued.

The Cryhocycle was a machine that appeared to hold promise for eliminating the radiators. It appeared that at the time when the orbiter contained large quantities of oxygen and hydrogen, that a Cryhocycle which was able to produce power and simultaneously provide cooling, would provide an overall weight savings to the orbiter. When the cryogenically stored propellant used for the OMPS and ACPS was removed from the vehicle, some of the advantages of the Cryhocycle were lost. Indeed, a comparison study between a baseline system and a Cryhocycle System for the current orbiter shows the Cryhocycle to have a weight disadvantage. The baseline system was assumed to consist of fuel cells, radiators, and dedicated hydrogen for cooling during reentry, along with the associated cryogenic storage tanks. The

Cryhocycle System consists of the Cryhocycle (a hydrogen expansion machine which drives an alternator) and supplemental hydrogen for reentry cooling, along with the associated cryogenic storage system. (Studies by Grumman indicate the systems are nearly equal in weight). It appears that since there is no significant weight advantage and that the Cryhocycle is not developed as much as the fuel cells, it would be appropriate to place the development emphasis on the baseline system. Furthermore, it seems that technology effort would be better placed on oxygen/hydrogen APUs than on the Cryhocycle.

Studies of the methods that can be employed for EC/LSS cooling during periods when the radiators are stored show that hydrogen is the lightest system. It appears that a dedicated hydrogen system that provides cooling for both the EC/LSS and the APU is desirable.

The system can be operated from lift-off to orbit injection or until radiator deployment and cooldown. In preparation for descent, the system would be operated from radiator storage to activation of the ground cooling system.

During ferry flights the APUs should not be operated. The hydraulic and electrical power would be provided by jet engine power pads. The heat generated by these components would be rejected by a standard aircraft-type air/oil heat exchanger. The cooling for the EC/LSS could be supplied by a jet engine bleed air cycle machine that can be attached to the jet engine pads.

The hydrogen system would not be required for ferry or horizontal flight tests.

Of all the systems studied, a common element is a cryogenic heat exchanger. In particular, for those systems recommended, Freon 21/hyrogen, Freon 21/oxygen, and tube oil/hydrogen heat exchangers should be developed.